Alternatives to Compressive Cooling in Non-Residential Buildings to Reduce Primary Energy Consumption

Final Report

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1. Background

The research project “Alternative Cooling Strategies for Non-Residential Buildings to Reduce Primary Energy Consumption” is funded by the Deutsche Forschungsgemeinschaft (DFG) [1] and the U.S. Department of Energy and refers to a study (“Environmental Assessment of Low Energy Cooling for Buildings”) initiated and performed by the Lawrence Berkeley National Laboratory, Berkeley, California (LBNL), the Intep AG, Zürich, and the Eidgenössische Technische Hochschule Zürich (ETHZ).

The study to investigate the primary energy use and the environmental impact of buildings with low energy cooling systems was started in 1994. The first task was to design and to optimize a state-of-the-art building, which could be built and located either in Central-Europe (Germany/Switzerland) or in the USA (California). Due to the different climatic conditions and construction details, different types of windows and exterior walls had to be taken into account. The design and the construction data for the European type of building was provided by Intep [26], the Californian window and wall constructions were defined by LBNL considering common types. In a preliminary study, the impact of the different types of windows and exterior walls had been investigated for several climates by using the simulation program DOE-2E. A first study on the environmental impact of cooling systems for buildings was made by the ETHZ [30].

The basic description of the building and many of the findings from the first part of the low energy cooling study are used with some alterations for this DFG-project, which began in June 1995 and has been finished in May 1997. The investigation has been conducted in the Energy Performance of Buildings Group of the Indoor Environment Program of the Environmental Energy Technologies Division at Lawrence Berkeley National Laboratory, Berkeley, California.

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2. Introduction and Goal

Cooling and air-conditioning are mostly required in today's office buildings because of two major reasons. The first one is the building itself. The following mentions only some issues for needing an HVAC system in modern office buildings. The share of glass area has increased within the past decades, and effective devices against solar radiation were often not installed or not properly used. High thermal loads due to solar radiation are the consequence. In many cases, the building structure does not provide sufficient thermal storage capacity to reduce the peak temperatures, so that without any HVAC system in place, uncomfortable indoor air conditions are very likely to occur, even in moderate climates. Open-plan offices require mechanical ventilation, as natural ventilation is difficult to control and the outside air cannot always be supplied sufficiently to the inner spaces. If the building is situated in urban areas, outside air often is polluted and has to be filtered before being supplied to the spaces.

Secondly, the electrical power requirements in modern offices have been increasing dramatically, as modern office equipment (computers, printers, fax machines) is being used at almost every work place. Additionally, the lighting system installed usually does not offer the possibility to control the lights according to the actual demand. All these electrical devices cause high interior thermal loads, which are at least of the same magnitude as the exterior loads. The cooling load in modern, well-designed office buildings is normally more influenced by the interior heat sources than by the outside air temperature or the solar radiation (if shading devices are being used). Due to the interior loads, cooling of the core zone of a building can be necessary even in wintertime, while areas close to the exterior walls might have to be heated.

The careful use of energy resources will become even more important, as the modern communication and office equipment will expand in the near future and the demand for thermal comfort in air-conditioned spaces increases.

Low primary energy demands combined with comfortable thermal indoor air conditions can only be achieved if the building is adapted to the needs of the occupants and the climate and the HVAC system is designed to consider thermal comfort requirements, primary energy consumption and environmental impact. Engineers and architects have to work together in order to design new, or retrofit existing, office buildings fulfilling these needs.

Modern, well-designed HVAC systems can almost always provide comfortable indoor air conditions if the building was designed to consider the climatic conditions of the particular location and the intended usage of the spaces. Almost all HVAC systems use compressor-driven chillers, which are able to provide the required supply water temperature without any technical problems, if the size of the chiller is appropriate. Conventional compression chillers usually offer low first costs and are easy to operate, but the primary energy consumption for cooling and dehumidification can make up a major part of the buildings primary energy use. The electrical peak power demand of a compressor-driven chiller usually determines the building's power peak, therefore, the rates of electricity are strongly linked to the chiller's capacity. Most compression chillers still use refrigerants, which are harmful to our environment. Research results showing the destruction of the ozone layer of the atmosphere (CFC-problem) and the problem of global warming, accompanied by the general need to reduce the primary energy consumption, make it necessary to think about alternatives to conventional compressive cooling. Replacing or at least supporting the compression chiller by less energy intense methods can contribute to a reduction of building's primary energy use.

In the past years some interesting cooling strategies have been rediscovered, developed and investigated, e.g., night ventilation, evaporative cooling, desiccant cooling, but a comprehensive comparison in terms of primary energy use is not yet available. Alternative cooling strategies are more often limited regarding their cooling capacity, which will vary during the day and the season. Any comparison of an alternative cooling strategy with a compression chiller operation needs to consider both the annual characteristics and peak load conditions. If alternative cooling offers remarkable or even sufficient potentials under peak load conditions, the possibility to substitute or support the compressive chiller is more likely for less critical conditions. Using a powerful building energy tool makes it possible to evaluate annual energy consumption as well as peak power demands. For this purpose, weather data available in Test-Reference-Year files can be used as an important input for a simulation program.

This investigation was initiated to compare the performance of several alternative cooling strategies with a modern well-designed conventional HVAC system with compression chiller. The alternatives will not necessarily be able to replace the compressor-driven chiller completely, but could at least decrease the primary energy use and/or the power demand of the HVAC system. To evaluate both the limits and the advantages of the alternative cooling strategies, this investigation uses one office building and considers some different climates.

The final goal is to show realistic potentials of several alternative cooling strategies to reduce the primary energy consumption of the building and to substitute or at least support the conventional compression chiller.

3. Investigation Method

The investigation presented is based on dynamic computer simulations using the building simulation tool DOE-2, version ‘E’, which has been developed at LBL. DOE-2E is an hourly energy analysis program calculating the energy performance of a building and its HVAC system.

It basically consists of five parts[2], which are internally linked to each other :

1) The Building Description Language (**BDL**) provides all dimensions, coordinates and properties of the walls, windows, floors and ceilings.

2) The **LOADS** part contains all definitions concerning the interior thermal and electrical loads, e.g., schedules for lighting, number of persons, electrical input of office equipment, and information about outside air infiltration rates. The link to the weather data is established in this part, too. All these data are used to calculate the cooling and heating loads for the spaces by considering a fixed, user-defined indoor air temperature.

The loads calculations are used for the automatic sizing of the chiller, heater and fans, if the user does not fix the sizes of these components. As only one setpoint temperature for both cooling and heating is available, the calculations might not reflect exactly the intended indoor air conditions for the summer and winter season, but result in components, which can cool, vent and heat the building sufficiently. For optimizing the HVAC system, the user has to define at least the cooling and heating capacities and the supply airflow.

3) The type of HVAC system, its components, schedules, airflow rates and the setpoints for the air-conditioned zones are defined in the **SYSTEMS** part. It uses the results from the LOADS part and adjusts these according to floating indoor air temperatures and latent (interior and exterior) loads. Special cooling, heating or ventilation strategies (economizer, heat-recovery, night-ventilation), particular characteristics of the components, e.g., cooling coil performance, and several schedules (setpoints, fans on/off etc.) can be defined separately.

4) The **PLANTS** part combines all components of the system or all systems (if there are more than one) to one plant. The types and characteristics of chillers and heaters and other components, e.g., cooling tower, can be fixed, or self-sized.

The SYSTEMS and PLANTS parts offer the opportunity to optimize the HVAC system. The output of these parts provides the user with information needed to evaluate the performance and energy use of the building and the system under study.

5) The **ECONOMICS** part uses, e.g, user-defined energy rates and calculates prices for energy consumption and life-cycle costs.

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[2] Detailed descriptions of DOE-2E can be found in the manuals [51,52]

4. Weather Data Analysis

In order to learn about the specific characteristics of each of the selected cooling strategies, the building's energy performance is calculated with the building exposed to different climates. The weather data for each location is provided by the corresponding Test-Reference-Year (TRY)[3], which offers hourly data of the most important climatic parameters.

To decide which American and European weather files were representative, the following main parameters influencing the cooling load and the performance of a HVAC system were analyzed :

- dry-bulb temperature
- humidity (moisture content, dewpoint and wet-bulb temperature)
- global solar-radiation

Eventually, the following climates were chosen for this investigation :

1) German TRY 1 (German coast, Bremerhaven, **Kiel*)**)
2) German TRY 3 (Hamburg, Essen, Bremen, **Berlin*)**)
3) TRY for the Lago Maggiore, Switzerland (**Locarno*)**, Magadino)
4) TMY for **Red Bluff*)**, Northern California
5) TMY for **San Francisco*)**, California

*) : The bold printed city names are used in the following text to distinguish the climate

4.1 Statistics on the entire year

Figure 4.1 through Figure 4.5 show statistics on the main weather parameters for the chosen climates. Tables presenting the mentioned weather parameters analyzed by number can be found in the appendix (Table A1 through Table A5).

Table 4.1 (page 14) summarizes the most important parameters by average values for the summer, winter and the whole year.

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[3] Two different types of weather files are being used :

a) Test-Reference-Year (TRY) : Real data base continously recorded over one year. The German TRYs are based on recordings in 3-hour-steps. The hourly data provided was generated by interpolation. The TRY represent the average characteristical climate for the corresponding region. Created hourly weather data, based on recordings between 1951 and 1970 (for Germany).

b) Typical Metrological Year (TMY) : Metrological data was hourly recorded over the period from January 1953 to December 1975. Each month is selected from the entire time span as the most representitive. Artificial data year with months from different years

<u>Dry-bulb temperature :</u>

The dry-bulb temperature influences the sensible load of the building due to heat conduction through exterior walls and windows and the sensible cooling capacity of the cooling system. The dry-bulb temperatures at night are important for the ability to cool a building with outside air during the night. Figure 4.1 shows the cumulative distributions of the dry-bulb temperature for the whole year. The chart's abscissa presents the annual hours in % which are below the corresponding temperature value (ordinate).

The two German climates do not differ very much from each other. Almost 80 % of the yearly hours range between 2 ^{o}C and 20 ^{o}C. The American climates under study are very different. Red Bluff shows a fairly high percentage (35 %) above 20 ^{o}C and only about 1 % below 0 ^{o}C. The dry-bulb temperature in San Francisco happens to be within a very narrow range. About 60 % of the annual hours are between 10 ^{o}C and 16 ^{o}C, there are no hours below 0 ^{o}C. The dry-bulb temperature values for Locarno are ranging between the German data and the values for Red Bluff. There are approximately 9 % below 0 ^{o}C and 17 % above 20 ^{o}C.

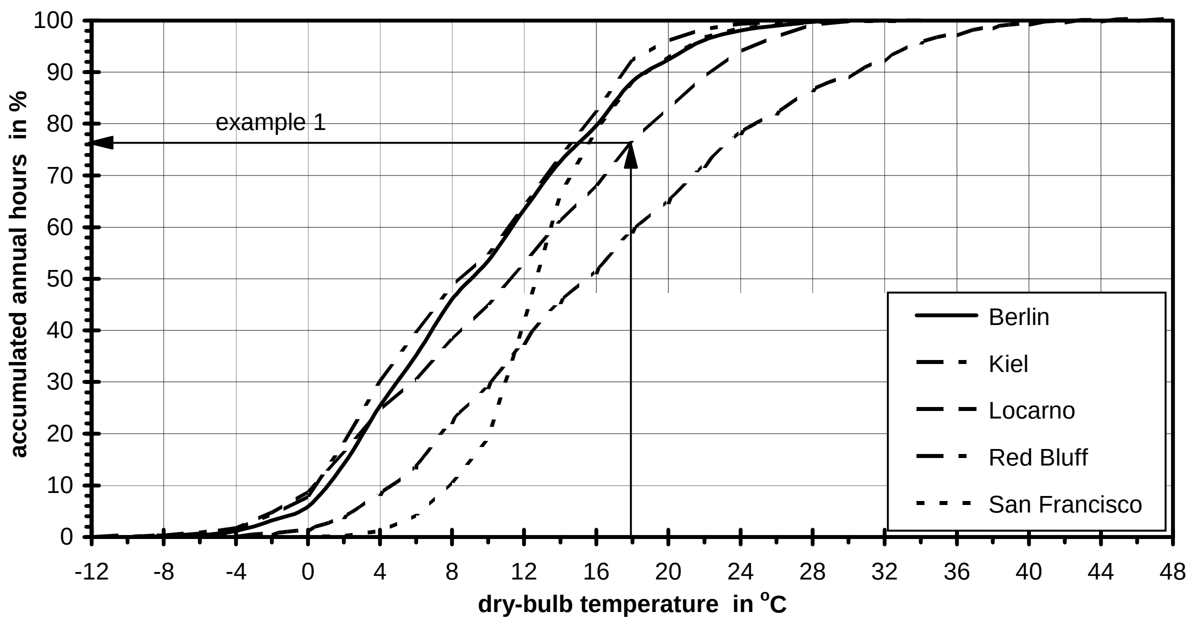

Figure 4.1: Cumulative distributions of the dry-bulb temperature for the five different climates investigated.
Example 1 : The dry-bulb temperatures in Locarno are below 18 ^{o}C at 76 % of the entire year; 24 % are warmer than 18 ^{o}C, respectively.

Humidity ratio :

In combination with the dry-bulb temperature, this basic weather parameter defines both relative humidity affecting thermal comfort and the wet-bulb temperature determining the exterior latent cooling load due to dehumidification. The dewpoint temperature fixing the required minimum coolant temperature for dehumidification depends on the humidity ratio as well. Figure 4.2 shows the percentage of hours of the year below a certain humidity ratio of the ambient.

The Californian climates are fairly dry locations[4]. The moisture content in Red Bluff exceeds 10 g/kg at only about 4 % of the year. Only about 1 % of the annual hours are above 10 g/kg in San Francisco. The Swiss climate Locarno offers many more humid hours with about 18 % above 10 g/kg and about 5 % even above 12 g/kg. The curves for the two German climates are almost identical in the upper humidity values and the humidity ratios are ranging between Locarno and the Californian climates. About 9 % of the hours in TRY 1 and TRY 3 are above 10 g/kg and about 1 % is above 12 g/kg.

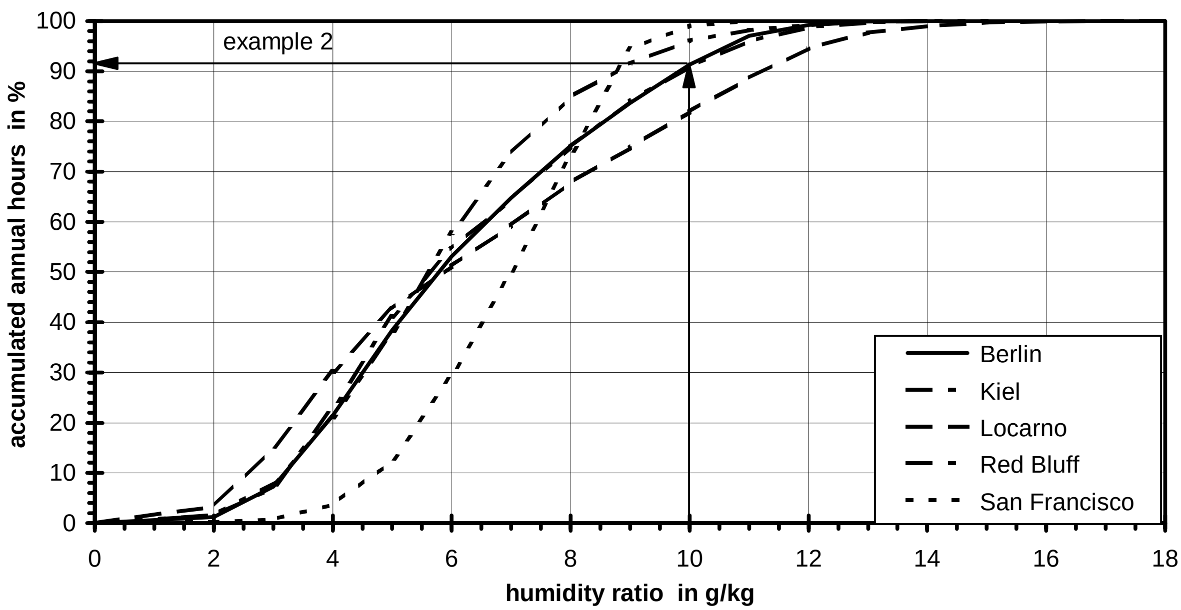

Figure 4.2 : Cumulative distributions of the humidity ratio for the five different climates investigated.
Example 2 : About 91 % of hours of the year in the German climates TRY 3 and TRY 1 have a lower humidity ratio than 10 g/kg dry air.

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[4] According to DIN 1946, part 2 [15], the recommended maximum humidity ratio in air-conditioned spaces is 11.5 g/kg (this corresponds to about 60 % relative humidity when the indoor air temperature is 26 ^{o}C) . Considering a certain internal latent load (people), the supply air humidity ratio should usually not be higher than about 10 g/kg.

Wet-bulb and dewpoint temperature :

The performance of the alternative cooling strategies investigated (water-based economizer, evaporative cooling, desiccant cooling) highly depends on the moisture related parameters of the outside air. The wet-bulb temperature limits the lowest dry-bulb temperature possible in a cooling tower or an evaporative cooler. An important parameter characterizing the requirements for relative humidity control in the spaces, the dewpoint temperature determines the highest cooling coil temperature still dehumidifying the supply air.

There is another special issue related to the dewpoint temperature. When hydronic-cooled ceilings are used for cooling the spaces, the indoor air dewpoint restricts the lowest supply water temperature determining the cooled ceiling capacity, as condensation at the cooling water pipes must not happen. In buildings with operable windows or when there is no dehumidification of the supply air, sensors registering the outside air dewpoint temperature are used to turn off the cooled ceiling when the dewpoint temperature approaches the supply-water temperature.

Cumulative distributions of the wet-bulb temperature (Figure 4.3) and the dewpoint temperature (Figure 4.4) are presented in the following.

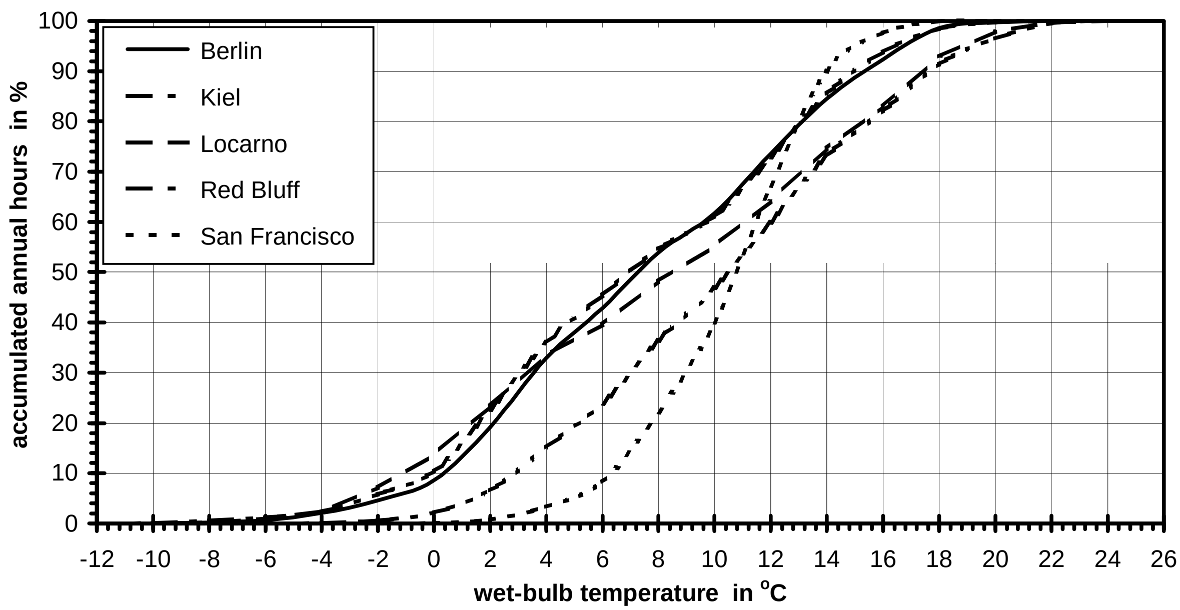

Figure 4.3 : Cumulative distributions of the wet-bulb temperature for the five different climates investigated

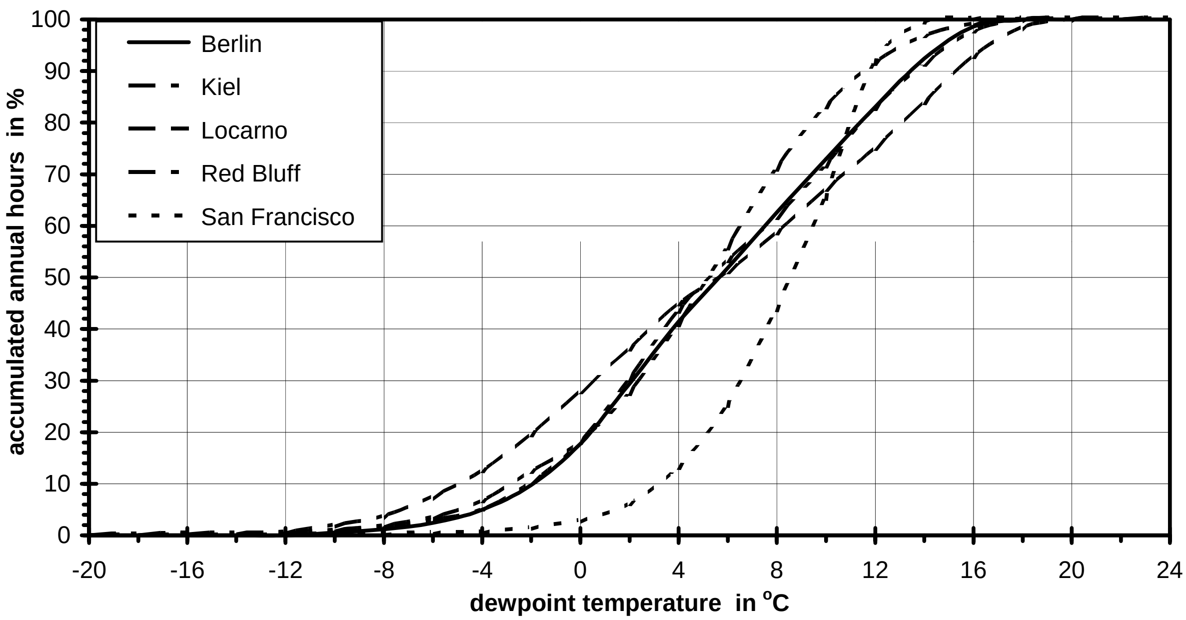

Figure 4.4 : Cumulative distributions of the dewpoint temperature for five different climates

Additional statistics on the humidity ratio and the wet-bulb temperature are presented in Chapter 4.2.

Solar radiation :

Solar radiation has a very strong influence on the sensible building load, especially with respect to shading devices. The highest solar radiation values usually determine the maximum exterior cooling load and, as a consequence, often the chiller capacity. However, the annual cooling energy consumption of a building is influenced by the amount of solar energy penetrating the building during the entire year. This value is represented by the annual sum of the global solar radiation $\sum I_{a,global}$ showing the possible annual solar gain (see Table 4.1).

The global solar radiation for the five different locations is statistically analyzed in Figure 4.5. The curves represent the daytime hours, i.e., while sun is shining ($I_{global} > 0$ W/m^2), which are below the corresponding global solar radiation value.

Figure 4.5 points out that the global solar radiation at the German locations is fairly similar and there are more than 60 % of the daytime hours below 200 W/m^2. Less than 10 % of the year provide a solar radiation above 600 W/m^2. Both Californian climates provide almost the same statistical curve of the solar radiation. Just about 34 % of the annual daytime hours have values below 200 W/m^2, but almost 30 % are above 600 W/m^2. The Swiss climate offers solar radiation values between those of the German and Californian weather files.

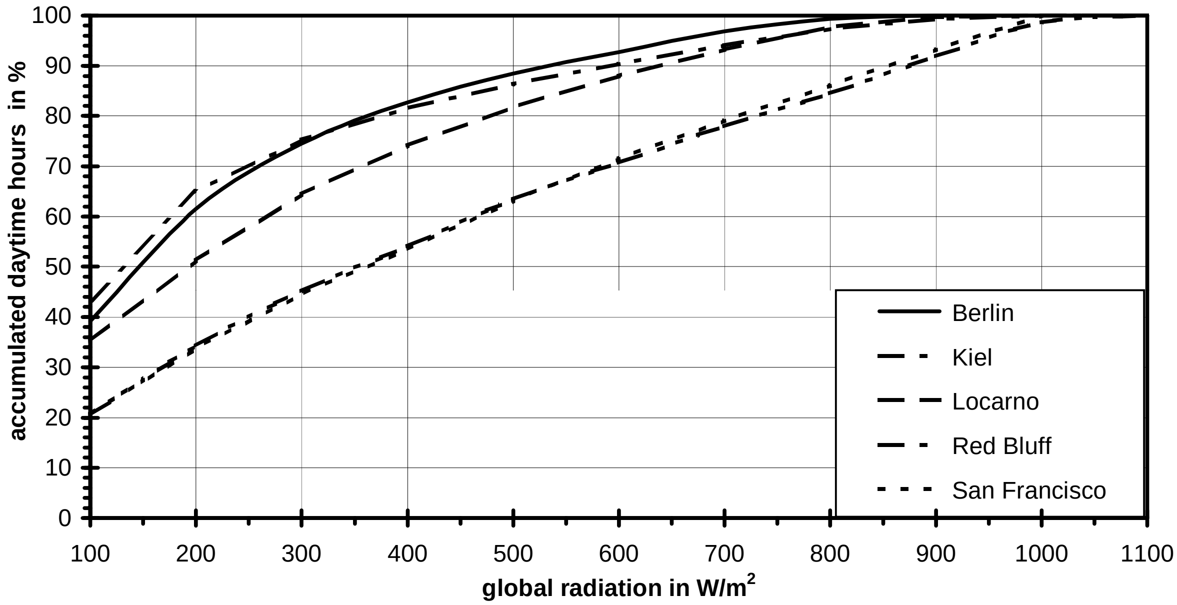

Figure 4.5 : Cumulative distributions of the global solar radiation for the five different climates investigated (only daytime hours).

Table 4.1 shows a summary of the annual average values for the weather parameters dry-bulb temperature $\bar{t}_{o,dry}$, wet-bulb temperature, $\bar{t}_{o,wet}$, dewpoint temperature, $\bar{t}_{o,dew}$, humidity ratio and global solar radiation for the five climates chosen. Additionally the mean values are shown for the cooling season (April - October) and the heating season (November - March). The annual mean value of the global solar radiation , $\bar{I}_{a,global}$, considers only the hours of the year with values above 0 W/m^2, i.e., daytime hours.

Table 4.1 : Average values of the dry-bulb, wet-bulb and dewpoint temperature as well as of the humidity ratio and the global solar radiation for the five climates investigated

average values		location				
		Berlin	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
$\bar{t}_{o,dry}$ in ^oC	winter*)	3.4	2.6	3.6	8.8	10.4
	summer**)	12.7	12.7	16.4	22.1	14.9
	year	9.5	8.7	11.1	16.5	13.0
$\bar{t}_{o,wet}$ in ^oC	winter	2.2	1.7	1.5	5.7	8.3
	summer	11.1	11.0	13.0	13.6	11.8
	year	7.4	7.1	8.2	10.3	10.3
$\bar{t}_{o,dew}$ in ^oC	winter	0.7	0.5	-1.6	2.0	6.1
	summer	9.0	9.3	10.4	6.9	9.4
	year	5.6	5.7	5.5	4.9	8.0
x in g/kg	winter	4.1	4.0	3.5	4.6	6.0
	summer	7.5	7.6	8.4	6.6	7.4
	year	6.1	6.1	6.4	5.8	6.8
$\bar{I}_{a,global}$ in W/m^2 ***)		210	213	260	406	400
$\Sigma\ I_{a,global}$ in kWh/m^2		982	981	1180	1822	1809

*) from November 1st through March 31th
**) from April 1st through October 31th
***) only the daytime hours with values above 0 W/m^2 are considered

4.2 Statistics for the cooling season

To point out the impact of the weather parameters on the cooling capability of the alternative strategies additional analysis on the most determining climatic conditions during the cooling season was conducted additionally. With regard to the Californian climates, the cooling season was set to the period from April 1st through October 31th for all the climates investigated regardless of the fact that the heating season in Germany usually already starts in September and lasts until the end of April. Considering usual office hours (8^{oo} h - 17^{oo} h) and operation schedules for the HVAC system, some extended statistics are presented in the following Figures.

Figure 4.6 depicts distributions of the wet-bulb temperature representing the cooling season hours between 6^{oo} and 17^{oo} h. Table 4.2 shows the corresponding percentages of cooling season hours below a certain wet-bulb temperature for the five climates investigated. Figure 4.7 depicts distributions of the humidity ratio representing the cooling season hours between 6^{oo} and 17^{oo} h.

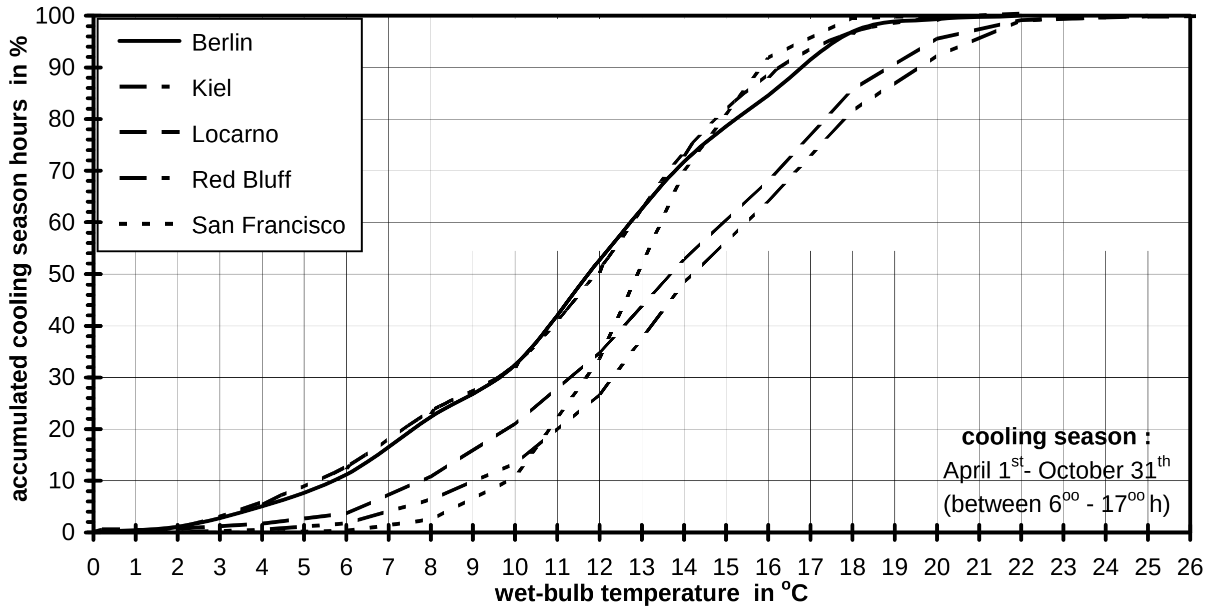

Figure 4.6 : Cumulative distributions of the wet-bulb temperature during the cooling season and between 6^{oo} and 17^{oo} h for the five different climates investigated (compare Table 4.2)

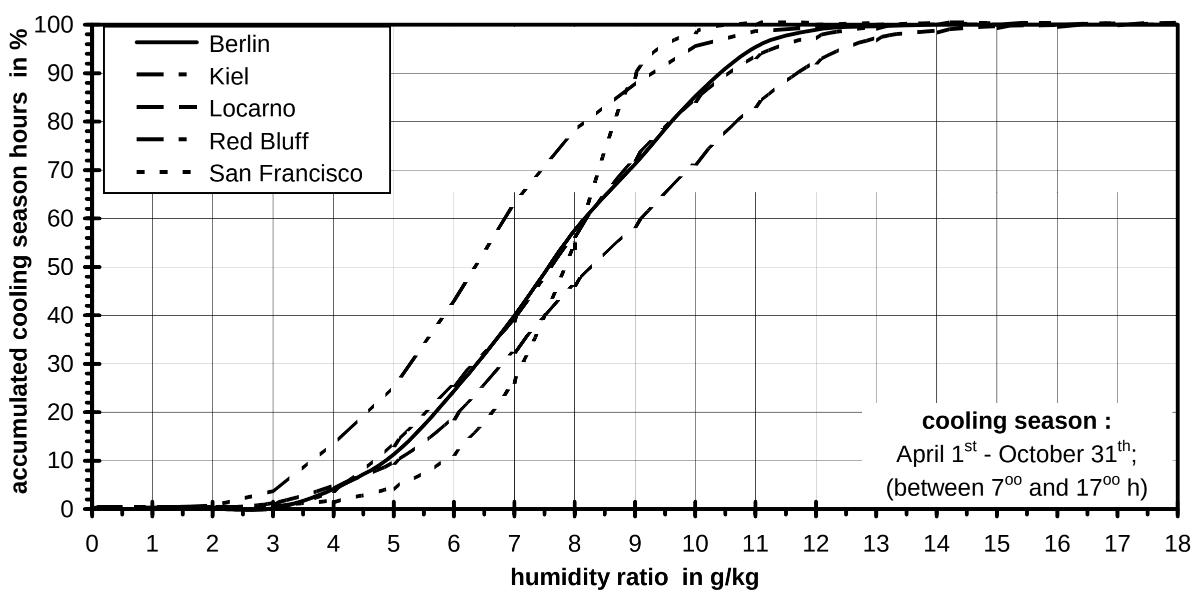

Figure 4.7 : Cumulative distributions of the humidity ratio during the cooling season and between 7^{oo} and 17^{oo} h for the five different climates investigated

Table 4.2 : Percentage of hours during fan operation time (6^{oo} - 17^{oo} h) in summer (April - October) below a certain wet-bulb temperature (compare **Error! Reference source not found.**)

wet-bulb temperature in ^oC	**location**				
	Berlin	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
0	0	0	0	0	0
5	6	8	2	1	0
10	32	32	21	13	11
12	53	50	35	27	34
14	72	73	53	48	70
16	85	88	68	64	92
18	97	96	86	82	100
20	99	99	96	92	-
25	100	100	100	100	-
$\bar{t}_{wet,day}$	11.4	11.4	13.5	14.3	12.8

example : 72% of the daytime hours during summer in Berlin are below 14^{o}C wet-bulb temperature

The following psychrometric chart (Figure 4.8) characterizes the particular climates by showing occurrences of dry-bulb and coincident wet-bulb temperatures during the daytime hours (from 7^{oo} h to 19^{oo} h). The corresponding values are presented in Table 4.3.

Detailed distributions of dry-bulb and coincident wet-bulb temperatures for all five climates are presented in Table A6 to Table A10 in the appendix.

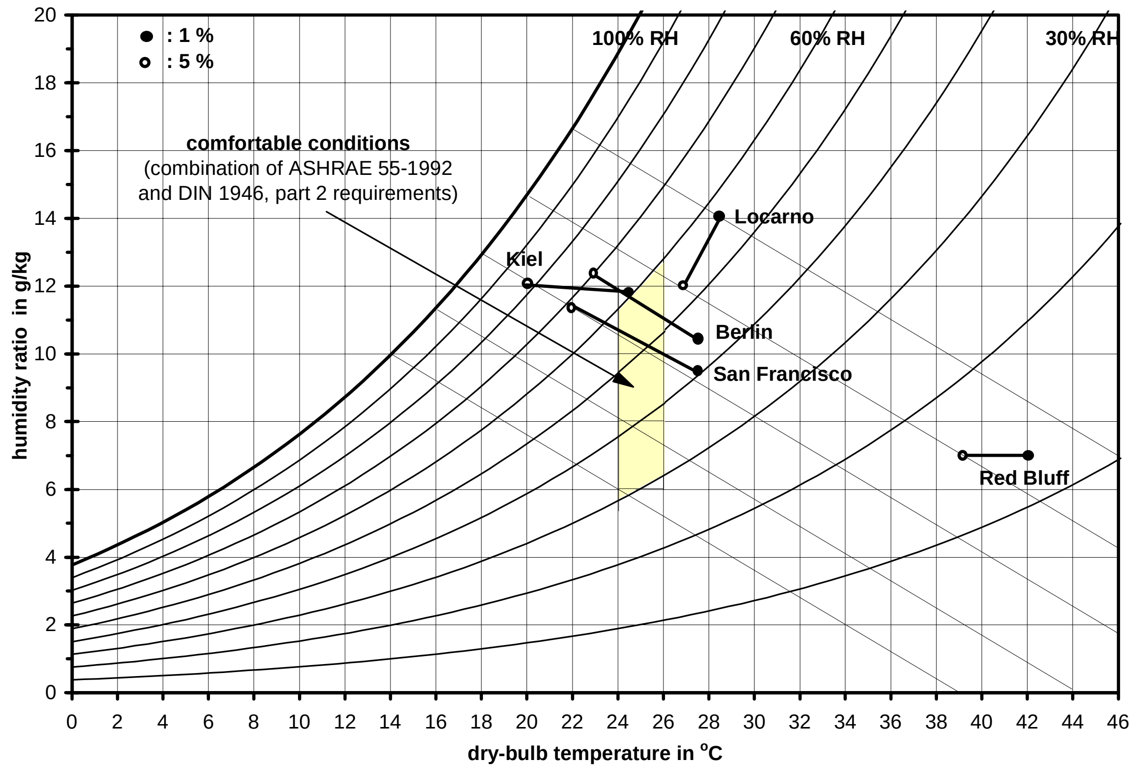

Figure 4.8 : Dry-bulb and coincident wet-bulb temperatures for the five climates investigated ("1 %" means that 1 % of the annual hours between 7^{oo} and 19^{oo} h exceed the shown dry-bulb and coincident wet-bulb temperature).

Table 4.3 : Characteristic climatic data for the climates investigated ("1 %" means that 1 % of the annual hours between 7^{oo} h and 19^{oo} h exceed the shown dry-bulb and coincident wet-bulb temperature; compare **Error! Reference source not found.**).

		location									
		Berlin		**Kiel**		**Locarno**		**Red Bluff**		**San Francisco**	
		1 %	5 %	1 %	5 %	1 %	5 %	1 %	5 %	1 %	5 %
dry-bulb	°C	27.5	23.0	24.5	20.0	28.5	27.0	42.0	38.5	27.5	22.0
wet-bulb	°C	19.0	19.0	19.0	18.0	22.0	20.0	21.0	20.0	18.0	18.0
dewpoint	°C	14.6	17.1	16.3	17.0	19.3	16.7	9.5	9.6	12.7	15.9
humidity	g/kg	10.6	12.4	11.8	12.4	14.3	12.1	7.5	7.6	9.4	11.6
RH	%	46	69	60	83	58	53	15	18	40	69

5. Energy statistics

This chapter presents a short statistical analysis of the energy consumption in the United States and Europe. Starting with general energy data, primary energy characteristics in office buildings will be presented and discussed briefly at the end of this chapter. Because of the climates chosen, the focus is on energy data for California, Germany and Switzerland. Additionally, charts for the United States are presented as well.

The entire primary energy consumption of California in 1992 was about 2100 TWh/a corresponding to a use per capita of 235 GJ/a. The United States consumed about 23000 TWh/a primary energy in 1992, which corresponds to 320 GJ/a per capita. The German primary energy consumption in 1992 was about 3950 TWh/a, or 177 GJ/a per capita. 70 % of the primary energy consumed in Germany needed to be imported to cover the entire demand [50]. Petroleum products made up about 70 % of the German primary energy imports [38]. Switzerland's primary energy use in 1992 amounted to 49 TWh/a, which corresponds to 143 GJ/a per capita. Having no energy resources but hydroelectric power, Switzerland had to import about 75 % to cover the Swiss primary energy consumption [38,50,54,55] . Figure 5.1 presents the primary energy consumption by energy sources in California and Germany[5].

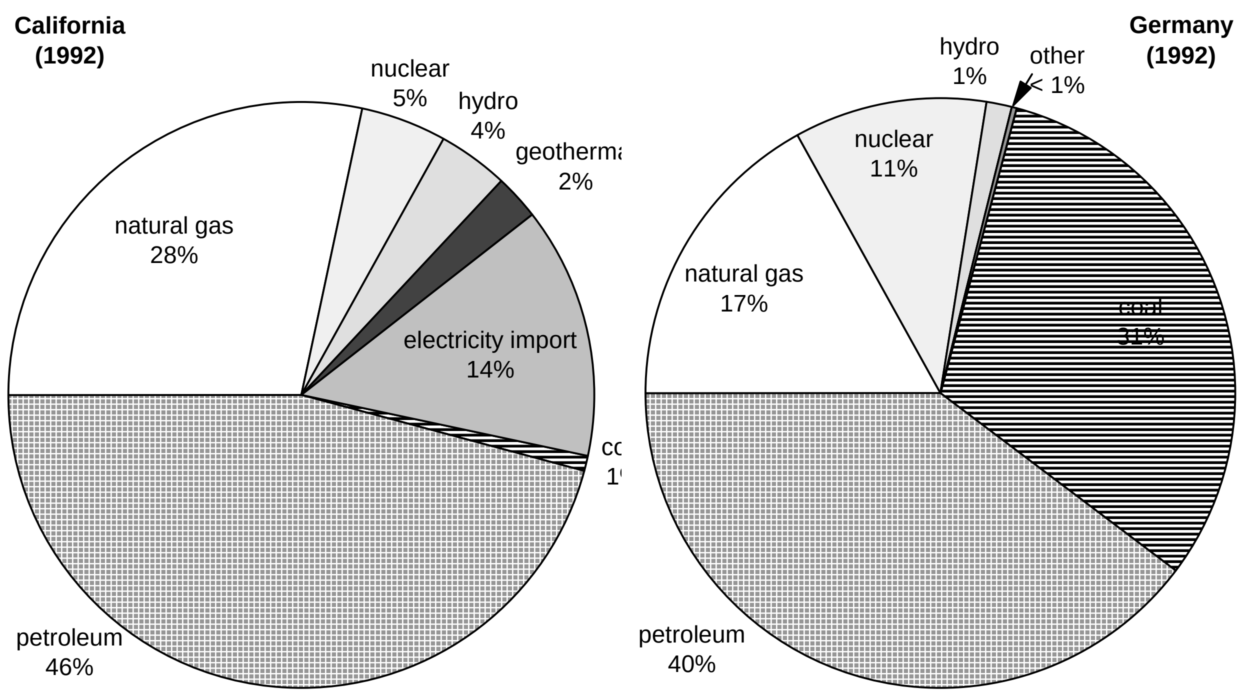

Figure 5.1 : Total energy consumption of Germany and California by primary energy sources [38,54].

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[5] The corresponding figures for Switzerland and the United States can be found in the appendix (Figure A1)

Figure 5.2 presents the primary energy use of California and Germany in 1991 by end-use sectors : transportation, industry, residential and commercial buildings. Commercial buildings demand the same percentage in both cases shown, which is also true for the United States (commercial buildings : 16 %, residential : 20 %, transportation : 27 %, industry : 36 %).

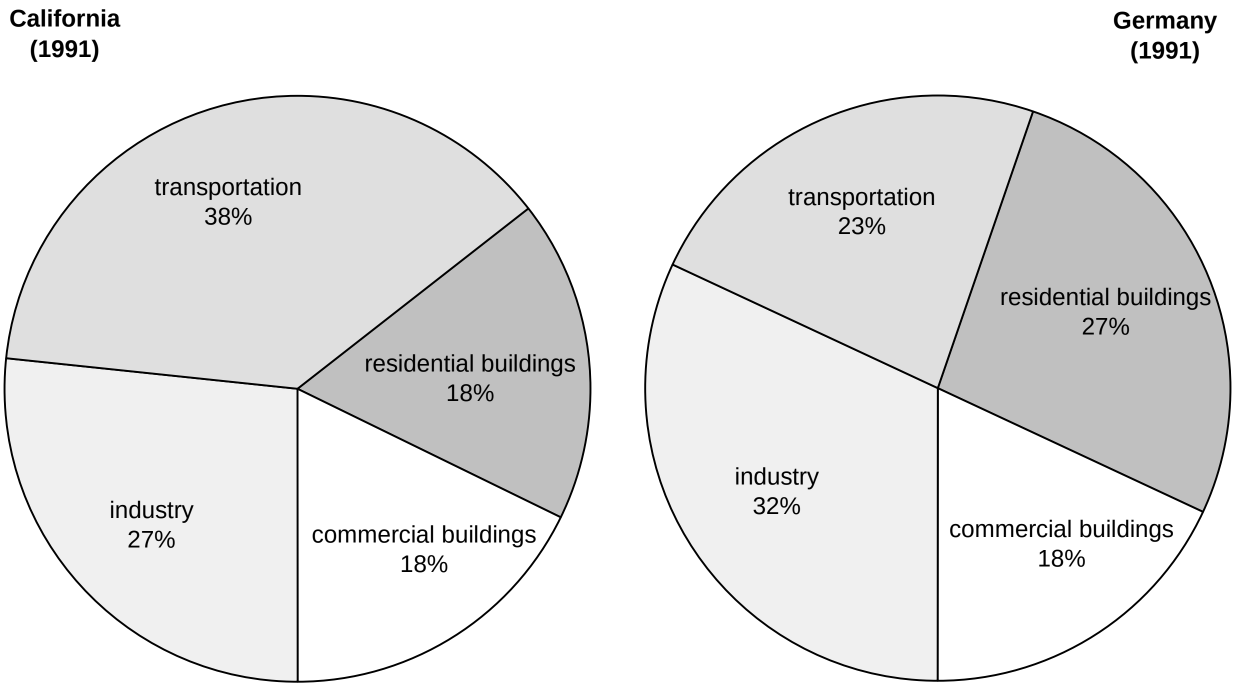

Figure 5.2 : Total energy consumption (including energy conversion losses) of Germany and California by end-use sectors [38,53].

Figure 5.3 shows the primary energy consumption of commercial buildings in Germany and the United States by energy sources. The category “electricity” contains both the site electricity and the conversion losses, thus, it represents the *primary* energy consumption. The significantly higher share of electricity consumed in commercial buildings in the United States is due to the fact that a lot of these buildings have to be air-conditioned and the climate in the United States usually demands more often air-conditioning than the German climate. Secondly, many heating devices in American HVAC systems are operated electrically to take advantage of the low first costs and the easy maintenance of electric appliances. This dramatically increases the electricity consumption and peak power demand. Apparently, American building owners, architects and engineers do not often consider the relatively high operation costs of electricity[6] and the impact on the primary energy sources (conversion losses).

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[6] Electricity is about 3 times more expensive per kWh than natural gas, when being used for heating a building either in California or Germany. The relation of the energy prices are more or less the same, but the energy prices in California are currently (1997) just about 50 % of the German prices for electricity or natural gas.

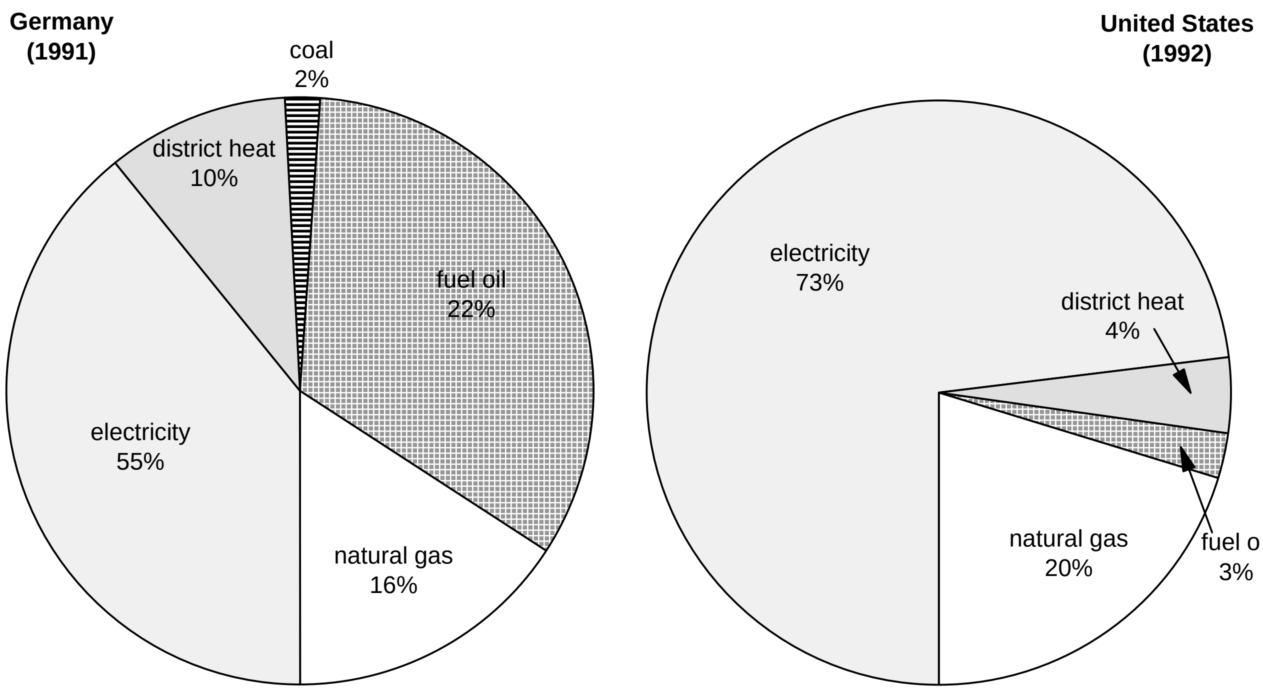

Figure 5.3 : Total energy use of commercial and residential buildings in Germany and the United States [55] in percent by energy source [38,55].

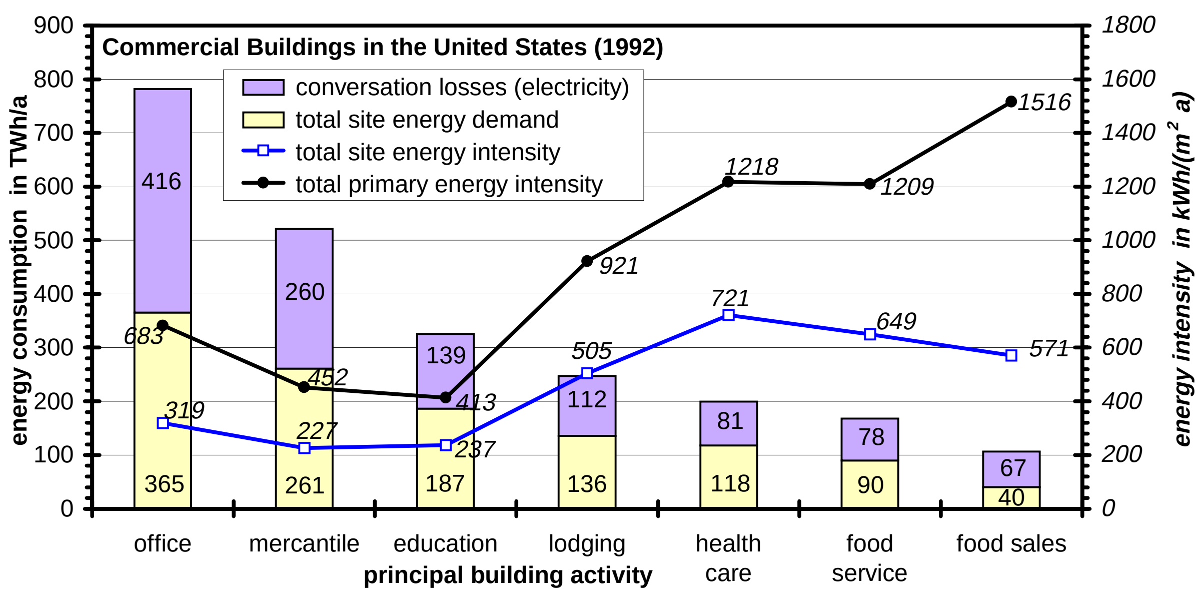

Figure 5.4 : Total primary energy consumption and primary energy intensity of American commercial buildings by principal building activity.

Figure 5.4[7] shows the results of an American survey recently published investigating the energy consumption of 4800 commercial buildings in the United States [55]. According to this survey, office buildings are using about 25 % of the primary energy consumption of American commercial buildings. The average primary energy intensity (including conversion losses occurring with the electricity generation) of American office buildings is about 680 kWh/(m^2 a).

Unfortunately, German commercial buildings have not been investigated as thoroughly as the American. As there is not sufficient data, the energy consumption can only be estimated using the few results available. These data [21,43] do usually not consider the conversion losses with electricity generation and sometimes do not even list the site electricity demand in detail. The site energy intensity of German buildings varies from 160 kWh/(m^2 a) (no cooling system) to 370 kWh/(m^2 a) (with HVAC system). An average German office building uses about 270 kWh/(m^2 a) site energy, which is only slightly less than the averaged total site energy intensity of American office buildings (319 kWh/(m^2 a); lower curve in Figure 5.4).

Considering the percentage of electricity used in office buildings (Figure 5.3) and the conversion losses, the *primary* energy intensity in German office buildings can be estimated to ragen between 220 kWh/(m^2 a) and 720 kWh/(m^2 a). The average for these buildings is about 400 kWh/(m^2 a).

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[7] Figure 5.4 does not present all principal building activities considered in the commercial buildings survey [55]. The shown categories represent about 60 % (by floor area) of all buildings investigated.

Figure 5.5 presents the total energy input to generate electrical power for Germany and California. Figures presenting the corresponding data for Switzerland and the United States can be found in the appendix (Figure A2).

More than 20 % of the entire electricity demand of California in 1994 was generated by several renewable energy sources. Switzerland uses hydroelectric plants to cover 60 % of its electricity consumed in 1993 (Figure A2), whereas Germany takes advantage of renewables for electricity generation to a very little extent (in 1991 : 4 %, with a trend to increase slightly).

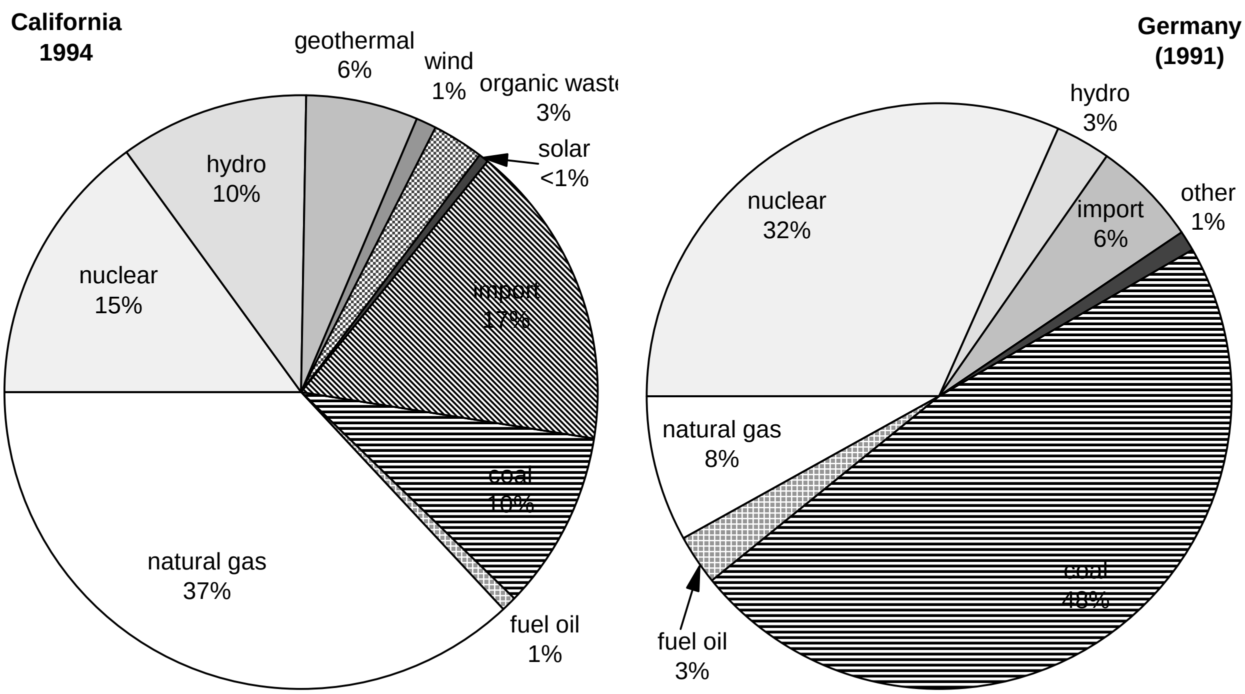

Figure 5.5 : Electricity generation in percent for California and Germany by primary energy source.

6. Outline of this investigation

The first step for comparing the efficiency of alternative cooling strategies with compressive cooling is to choose an appropriate building. This building has to be described in detail (the BDL of the DOE-2E input file). Also, the usage of all rooms and spaces has to be determined realistically by creating schedules for occupancy, lighting and interior thermal loads (LOADS in DOE-2E). The usage of shading devices and daylighting can be optimized by temporarily using a standard HVAC system, which needs not to be specially adjusted yet to achieve the most energy efficient operation. After the boundary conditions are chosen, the type of conventional air conditioning system can be selected and the particular devices can be sized (SYSTEMS and PLANTS in DOE-2E). Finally the alternative cooling strategies can be investigated and be compared with a HVAC system (reference system), which uses compressive cooling.

The simulation results with the conventional HVAC system for all five climates investigated are the basis for the entire project. These results provide all the required information about the conventional system, e.g., annual energy consumption, peak power demand, cooling load etc. All alterations with the air-conditioning system or its operating mode can easily be compared with the corresponding “reference HVAC system” and arising saving potentials become obvious.

The following alternative cooling methods are investigated:

- Night ventilation
- Evaporative cooling
- Desiccant cooling
- Absorption chiller

7. Description of the building

The office building selected for this investigation represents today's state-of-the-art design. This includes very low, but still realistic, heat transfer coefficients for the exterior walls and windows. Concrete floors and ceilings, which are not "covered" by dropped-ceilings or thick carpets, respectively, provide the building with a high thermal storage capacity. Day-lighting and exterior blinds controlled by solar radiation on the respective walls for each room reduce the interior and exterior cooling loads.

7.1 Building structure

Intep designed the building in collaboration with LBL [26]. The shape and construction of the building is based on an existing office building in Winterthur, Switzerland, with a favorable surface-to-volume ratio. The partitioning of the building is the same for all investigated climates.

For comparing the annual energy consumption of the different cooling strategies, one entire floor in the middle of the 5 story building was used[8] (Figure 7.1).

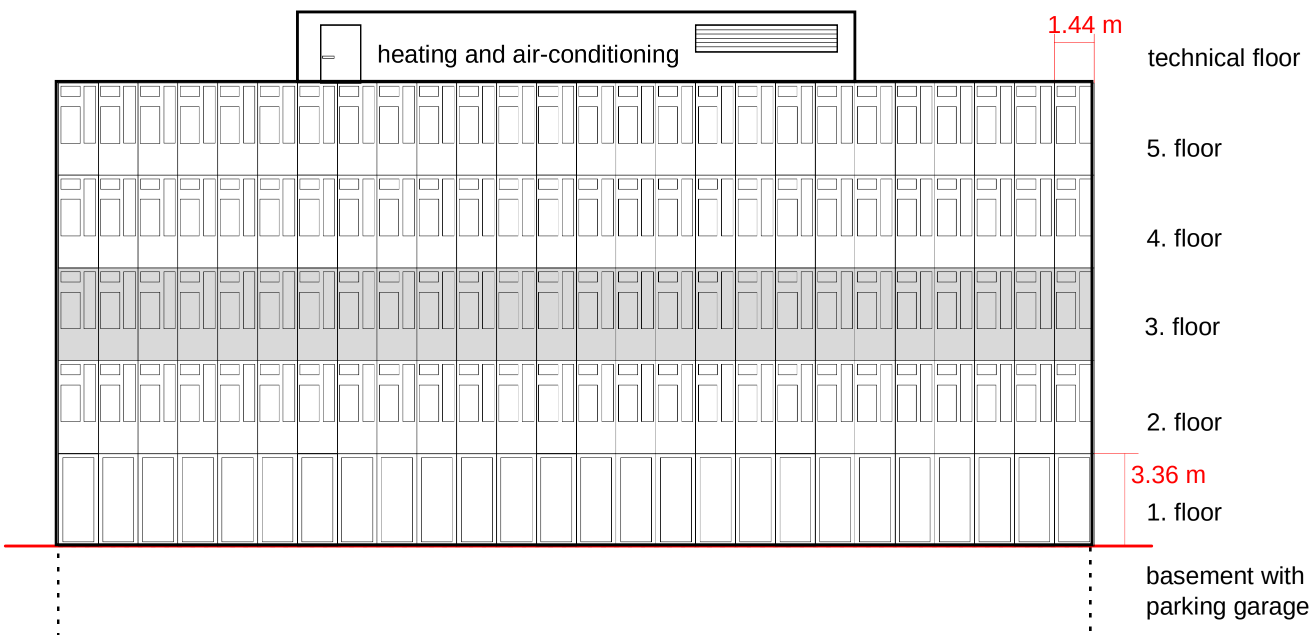

Figure 7.1 : View of the office building investigated (simplified window areas are used in the simulation runs; compare Chapter 17.4.1.2, appendix).

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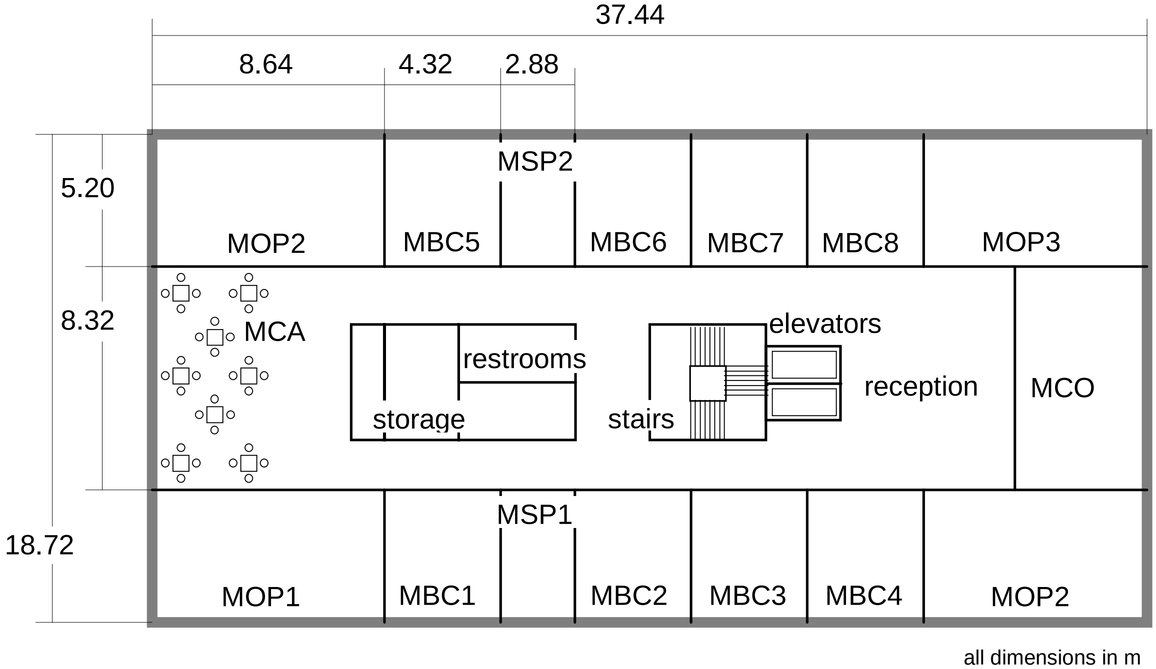
[8] As a middle floor usually has a lower energy intensity (kWh/m^2) than, i.e., the top floor with a similar usage, the energy intensity of the entire building will be slightly higher when compared with a middle floor.

7.1.1 Floor plan

The floor plans of the 2nd to the 5th floor are identical. Their partitioning is based on a grid of 1.44 m x 1.44 m. Figure 7.2 presents the floor plan of the investigated floor.



offices :	four persons (**O**pen-**P**lan)	: MOP1..MOP4 *)
	two persons (**B**ase-**C**ase)	: MBC1..MBC8
	one person (**S**ingle-**P**erson)	: MSP1, MSP2
	CAfeteria or break area	: MCA
	COnference room	: MCO

*) M stands for “**M**iddle” of the building

Figure 7.2 : Floor plan of the middle floors of the building.

7.1.2 Components of the building

The building is a steel-frame, the floors and ceilings are made of reinforced concrete with an embedded absorption layer for acoustic reasons.

For the European locations, the exterior wall-parts are made of reinforced concrete, which are integrated in the building’s construction. The wall elements are insulated and covered with wooden siding. The heat transfer coefficient (U-value) of these walls is 0.24 W/m^{2}K.[9] Window elements with a heat transfer coefficient of 1.58 W/m^{2}K (1.31 W/m^{2}K for the center of glass, 1.85 W/m^{2}K for the frame) are used. There are no columns between the window elements. The window's surfaces are set back against the vertical outline of the building by about 20 cm.

For the American locations, a curtain-wall construction, consisting of windows and opaque parts, is used. These elements are linked to the frame of the building with joints to the floors and ceilings. The opaque part of the curtain-wall is covered with a glazing element and provides a heat transfer coefficient of 0.44 W/m^{2}K. There is no set back of the windows.

Both the American and the European building are equipped with the same window element (Swiss construction). Previous investigations with different climates have shown that the characteristics of this window type has advantages when compared to some other windows with lower heat transfer coefficient, especially during the cooling period. The best insulating window types turned out to absorb much solar energy and transfer more heat to the spaces than the windows with worse U-value but no absorption characteristic.

In order to take advantage of the thermal storage of the concrete, neither the American nor the European building have dropped ceilings. Only the zones where the air ducts are placed are covered with panels. There are also no carpets covering the floors.

The most important information for the particular parts of the building are shown in Figure A3 and Figure A4 and Tables A11 to A14 in the appendix.

7.2 Interior loads

Each floor (2nd to the 5th) of the building is divided into a combination of offices with one, two or four work places (55 % of the entire floor area), a conference room (6 %) and a cafeteria (9 %). Restrooms, hallways and the elevators are located in the core zone of the building (30 %) (see Figure 7.2).

The working hours for the occupants of the building are set to 8^{oo} h until 17^{oo} h. The official American holidays are considered for all climates[10]. People being on their vacation are not considered. To determine the energy consumption, it is assumed that all employees and some visitors are in the building during the working hours. But not all offices are fully occupied during the working hours, as some occupants are meeting in the conference room and some are assumed of

[9] The heat transfer coefficient of an exterior surface is variable, because it depends among other parameters on the wind velocity and direction. DOE-2E uses hourly mean values for the heat transfer coefficient to calculate the heat gains or losses of a space. The indicated U-values are averaged values for the whole year.

[10] There are ten official holidays in the United States. Germany does not have an uniform regulation on holidays. Depending on the region, there are between 9 and 12 official holidays in Germany.

having a lunch or a coffee break in the cafeteria. The occupants in the conference room and the cafeteria are employees (up to 75 %) and visitors (up to 50 %). The building is not occupied on weekends or official holidays. Figure 7.3 presents the occupancy during the working hours.

The energy consumption as a result of domestic hot water use in the restrooms or the cafeteria is not taken into consideration.

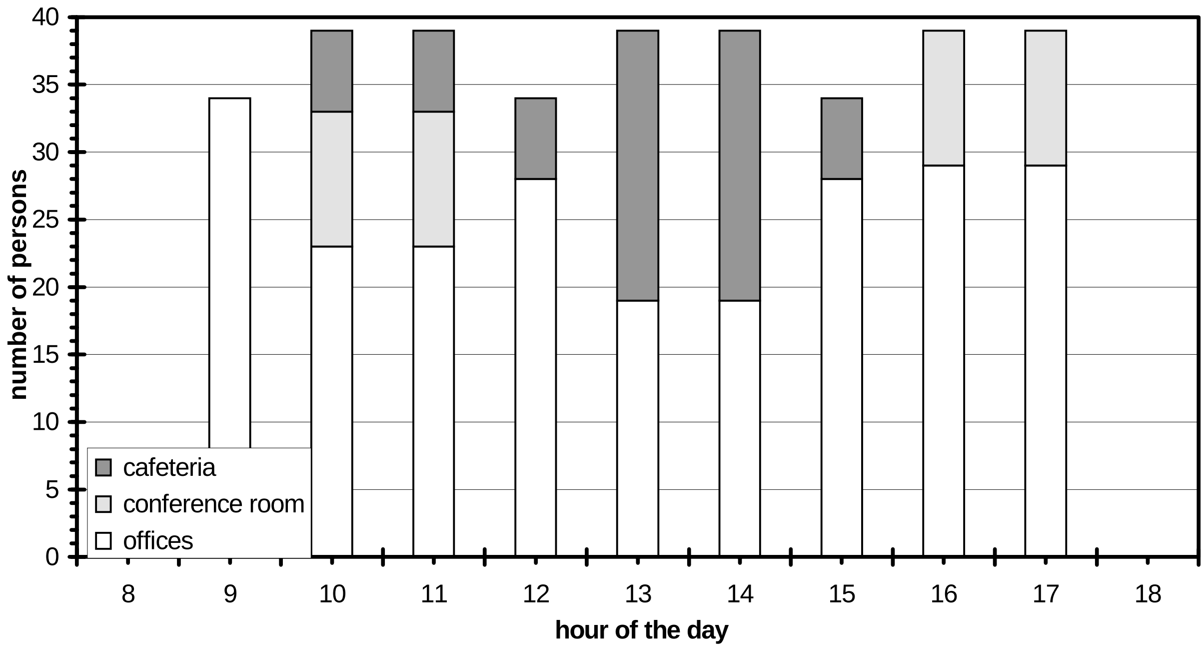


Figure 7.3 : Number of occupants during office hours[11] in the different spaces.

The offices are illuminated indirectly with the illuminance controlled automatically, by using both movement sensors and illuminance sensors. The required illuminance is fixed to 350 lx according to American and German experiences and standards [16,31]. The one- and two-person offices are equipped with one illuminance sensor, the four-person offices have two sensors. These illuminance sensors are located above the desks. The lights are stepped and have several lamps which can be switched on or off independently.

Each work place is provided with a personal computer and each office room is equipped with a printer.

The lighting in the conference room consists of indirect lights at the podium and several spots for illuminating pictures and drawings on the interior walls.

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[11] The *'hour of the day'* does not represent the *time of day*. 9 on the ordinate means the 9^{th} hour of the day, which begins at 8:01 am and lasts until 9:00 am. This definition is being used in DOE-2E [51] as well as for all following schedules and charts. According to this definition, the occupation hours are between 9 and 17, meaning from 8:01 am to 5:00 pm.

Illuminance-controlled lights light each of the 8 tables in the cafeteria. The hallways are lit constantly during the working hours, even on weekends.

The interior heat input for one entire floor is summarized in Table A15 of the appendix.

The setpoint of the room temperature[12] in all occupied spaces is chosen according to the requirements of DIN1946 [15]. In order to save cooling and fan energy, the highest permissible temperatures are allowed (see Figure 7.4).

As DOE-2E can only calculate and consider air temperatures, the indoor air temperature has to be estimated according to the required room temperature. In cooling mode the air temperature in rooms with all-air systems is usually lower than the room temperature, because of the higher radiant temperature of the surrounding surfaces, e.g., ceiling, windows, heat sources. In order to provide the required room temperature, the setpoint of the indoor air temperature should be at least 1 K lower [6]. The profiles of the required room temperature and the setpoint temperature for DOE-2E are presented in **Error! Reference source not found.**, which also shows the permissible range of the room temperature according to DIN 1946 [15] .

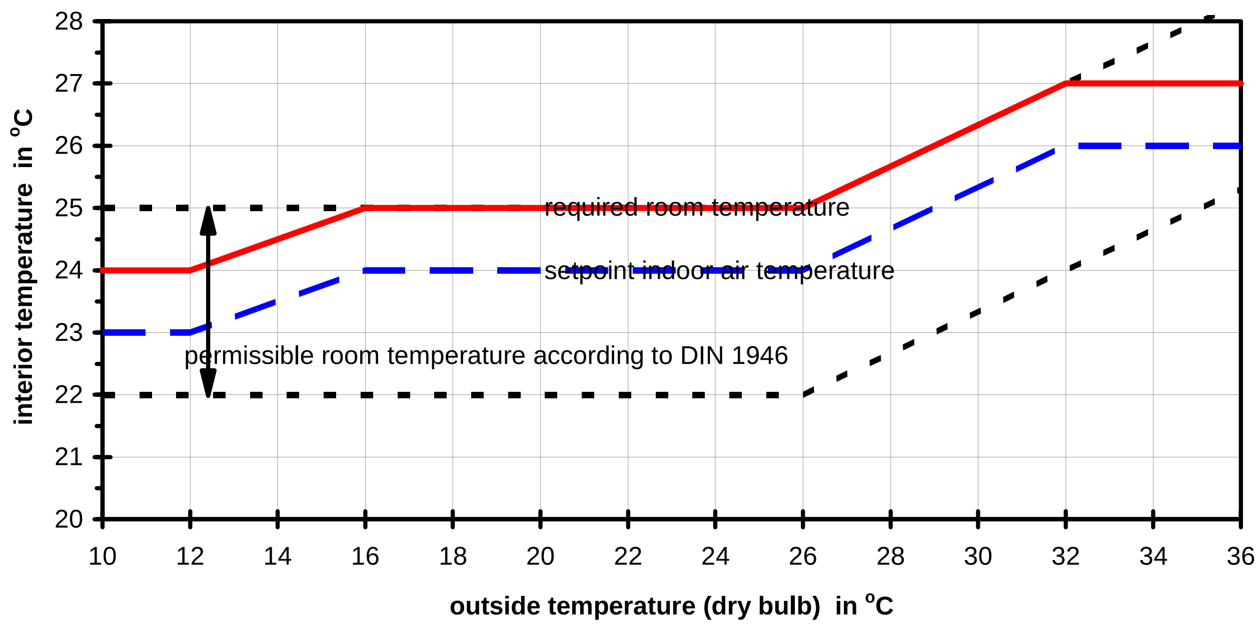

Figure 7.4 : Permissible room temperatures according to DIN 1946 [15] and setpoint air temperature for the offices.

[12] The room temperature or operative temperature is the average of the mean air temperature and the mean radiant temperature of a space and considers the heat exchange via convection and radiation [15,37]. This temperature is commonly used to determine thermal comfort.

8. Conventional reference HVAC system

Due to the external loads (solar radiation, outside air temperature) and internal thermal loads (people, lighting, equipment) and due to the demand on thermal comfort and indoor air quality, the building under study has to be thermally conditioned. The tasks for the air-conditioning system for this study are cooling the building, i.e., providing comfortable air temperatures and relative humidity levels, and supplying the rooms with the required outside airflow.

8.1 VAV system with compression chiller

The starting point for all comparisons is a conventional, but modern all-air system, which is supposed to represent the state-of-the-art in air-conditioning. For this purpose a variable-air-volume system (VAV system) was chosen. A water-cooled compression chiller provides the cooling with an electric input ratio of 0.33[13]. The heat removed from the building is rejected to the ambient by utilizing a cooling tower (see Chapter 8.6).

The supply air temperature is held constant during the cooling period (18 ^oC) unless dehumidification is needed. To dehumidify the supply air the air temperature is reduced at the cooling coils to about 14 ^oC. Adjusting the amount of supply airflow provides the required cooling capacity. The adjustment is realized by a speed-controlled supply fan and the VAV control units (VAV box) in every room. The supply air is delivered into the room by regular registers or diffuser is located at the ceiling and is mixed with the room air. The return air openings are located at the ceiling, too.

The VAV system is equipped with a heat exchanger (heat exchanger effectiveness: 0.70) to recover heat from the return air in wintertime. During the heating season the supply airflow is reduced to the minimum required for hygienical reasons, so that only outside air is delivered to the spaces with a flow rate of 10 l/s per person (maximum occupancy). The minimum outside air flowrate of the entire system is determined by the sum of the fresh air requirements for each of the spaces. The temperature of the supplied air is set to 21 ^oC. The rooms in the building are mainly heated by hydronic baseboard heaters, which are located at the exterior walls underneath the windows. A gas-fired water boiler

[13] Modern compressor-driven chillers achieve coefficients of performance (COP) of about 4.0, corresponding to an electric input ratio of 0.25.

The low COP-value of 3.0 has been used for this investigation by accident. Unfortunately, the “mistake” became apparent when it was to late to re-run all variations with compression chiller. However, all variations with compressive cooling (stand-alone or in combination with another strategy) can be compared without any disadvantage. The final discussion (Chapter 13) presents also results considering a COP-value of 4.0 for the conventional VAV system. The Figures shown allow evaluating the savings potential of the alternatives without any compressive cooling.

provides the hot water for the heating coils and the baseboards with an efficiency of 80 %[14]. Figure 8.1 shows a schematic for the modeled VAV system.

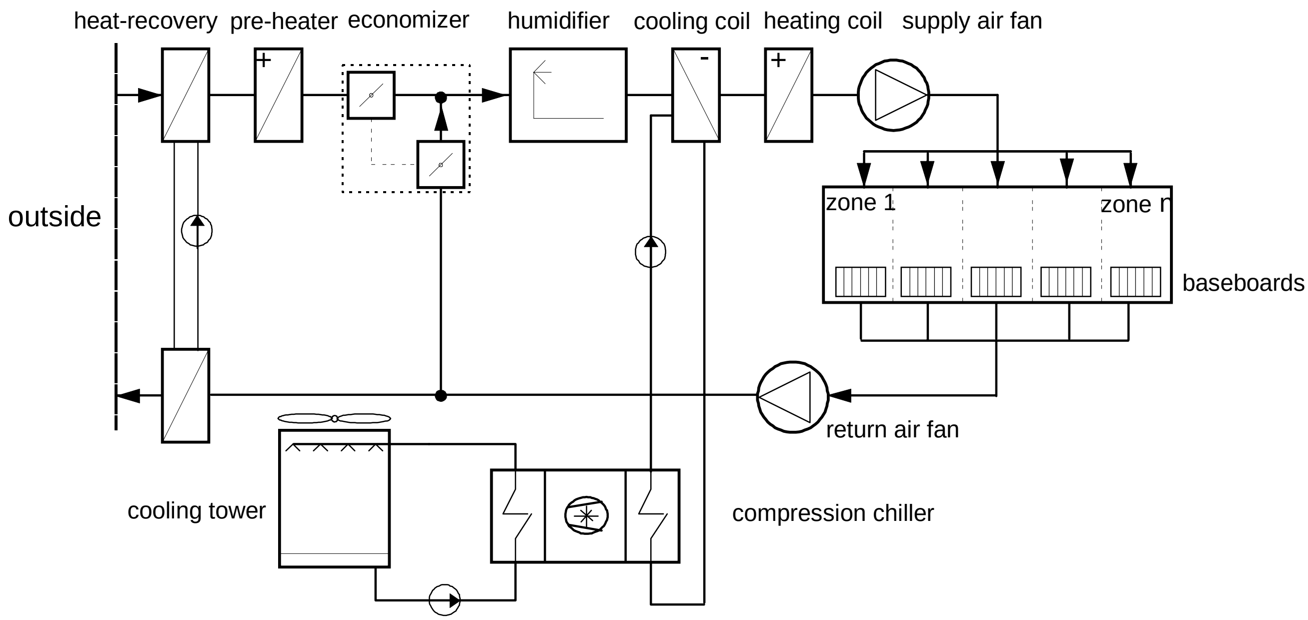

Figure 8.1 : Components of modeled conventional VAV system in DOE-2E.

Intep conducted a preliminary duct design for the conventional VAV system. Typical HVAC components were considered and design airflows determined by the respective loads were used. The calculations resulted in design pressure drops of about 700 Pa for the supply duct and about 400 Pa for the return duct of the reference VAV system. These values include the HVAC unit[15]. DOE-2E adjusts the pressure drop hourly considering the actual airflow ($\Delta p \sim V^2$).

The conventional HVAC system is not optimized yet for the first studies described in the following Chapters 8.2 and 8.3.

8.2 Variation of the building orientation

The solar gain often makes up the major part of the exterior thermal load. As the solar gain through the exterior walls is very much influenced by the angle between the solar azimuth and the azimuth of the largest exterior surface of the building and as the building's floor plan has a rectangular shape, the energy consumption to

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[14] This is the performance of a hot water boiler in DOE-2E by default. The boiler's efficiency is adjusted according to the current part load. Modern gas-fired boilers achieve an effectiveness of 0.9 to 0.95, However, this characteristic was not adjusted according to recent standards, as the performance in summertime has been the major point of interest.

[15] The detailed pressure drops of all components are presented in the appendix (Table A16).

thermally condition the building is dependent on the entire orientation of the building.

In order to determine the building's azimuth with the highest energy use, which is the most disadvantageous case for the alternative cooling methods, the entire building is modeled with different azimuths. As the parallel exterior sides of the building are identical and the rooms are almost equally used, three turning steps of 0^o, 45^o and 90^o are sufficient for this variation (Figure 8.2).

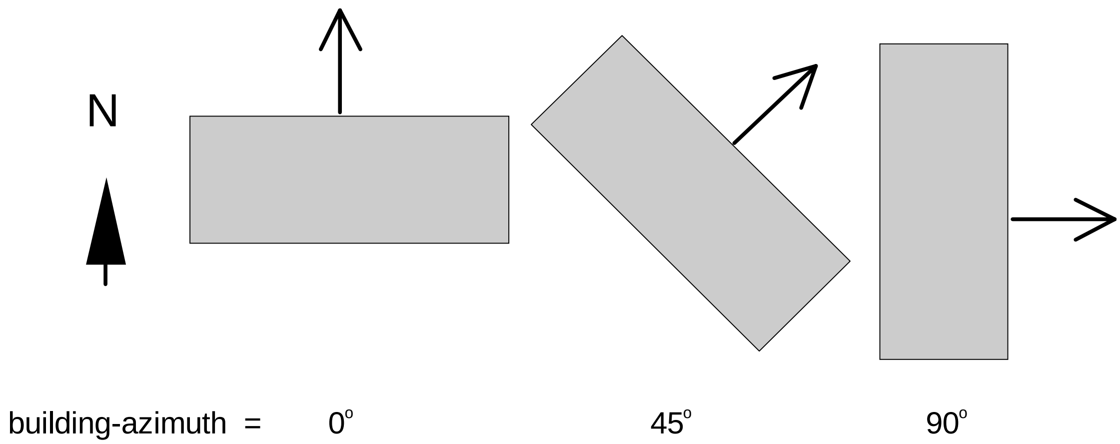

Figure 8.2 : Investigated building orientations.

Figure 8.3 shows the particular shares of the annual specific energy consumption (related to the floor area [60]) of the building for Berlin (German Test-Reference-Year 3) as a function of the building azimuth. For these examples there is no shading device being operated. The primary energy consumption presented takes into account that the electric energy for the chillers, fans, pumps, lights and equipment needs to be generated, i.e., conversion losses are considered. An efficiency of the power plant of 35 % is assumed for the entire investigation.

The total primary energy use[16] includes all categories and is almost independent of the building's azimuth, although it increases slightly by 2 % from 0^o to 90^o. The building's average peak cooling load and the annual cooling energy use have their maxima with an azimuth of 90^o. But in the room with the highest thermal load

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[16] The primary energy use shown in Figure 8.3 does not meet the requirements for an optimized building with a well-adjusted VAV system. Several parameters were fixed for this comparison, others (occupancy, time schedules for lighting or equipment load) were not set as they will most likely be found in reality, yet. An additional cooling energy demand for dehumidification is not considered either.

For the comparison these simplifications are of no disadvantage, but the shown values might not be taken for comparison with other results, presented in later chapters.

(which has two exterior surfaces) the highest cooling load appears at an azimuth of 45°.

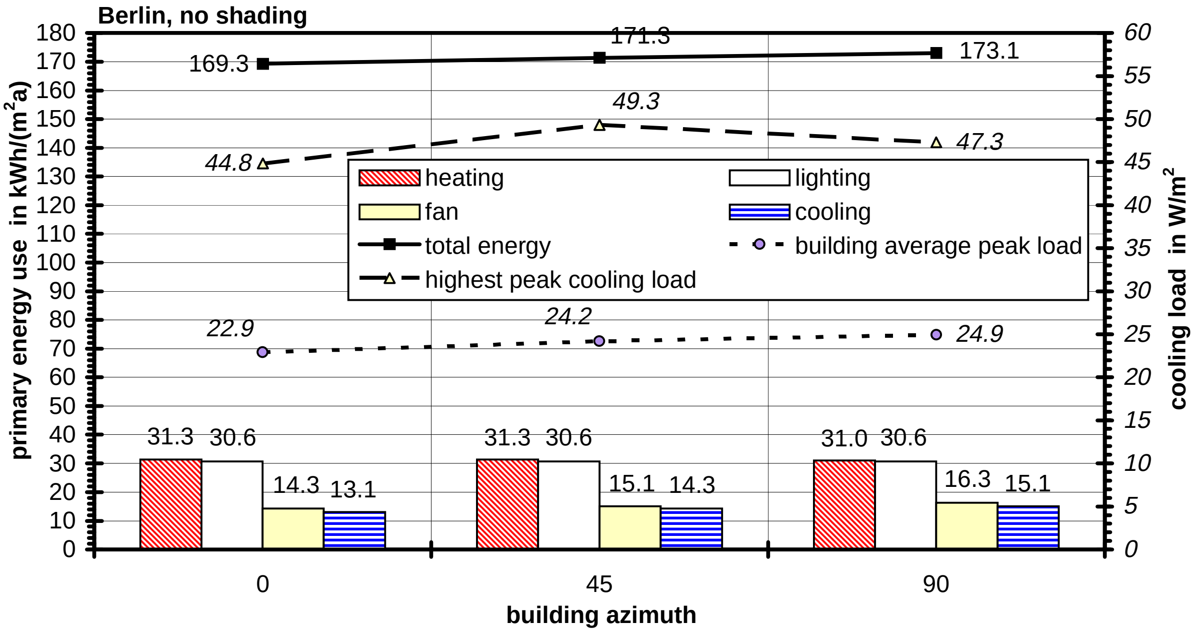

Figure 8.3 : Impact of the building orientation (azimuth) on the components of the annual energy consumption in kWh/(m^2 a) (without shadings; see footnote 16 on page 32).

The efficiency of an alternative cooling strategy is very much influenced by the highest expected specific cooling load, because this value determines the lowest coolant temperature, which represents the highest demand for the cooling device and the worst boundary condition for the cooling strategy. As no different characteristics are expected for different climates, a building azimuth of 45° is used for all further simulations. This means, the longer sides of the building are facing north-east and south-west, respectively.

8.3 Optimization of the shading control

Exterior blinds[17] are the most effective method to control solar gains through the windows. But closed blinds simultaneously reduce the light levels in a room, and a higher electrical demand for lighting could become necessary. As a consequence, the cooling load and the energy use for lights and cooling might be higher than in a similar room without blinds. An automatic control device for the blinds can provide both reduced solar gains and sufficient light levels, if the critical value for the solar radiation (shading trigger) is optimized. The best shading trigger value can be determined by several simulation runs using different threshold values for the

[17] The Venetian blinds used in this investigation are either closed or opened, the angle of the blinds is not being controlled. A constant shading coefficient of 0.13 and a visible transmittance value of 0.35 are considered when the blinds are closed.

shading device. These automatic shading devices and controls require regular maintenance.

Although the cooling energy is the most important energy parameter for this investigation, the optimization of the shading control has to focus on the energy use due to lighting as well, since it is strongly influenced by the blinds and at least of the same order of magnitude as the cooling energy consumption.

Figure 8.4 shows the building’s primary energy use in % as a function of the shading trigger value for the five different climates investigated. The reference level of 100 % is set to a shading trigger value of 0 W/m^2 , corresponding to a shading device closed at all times. Apparently, the most energy efficient setpoint for the shading control is between 200 and 300 W/m^2. Due to the significantly higher solar radiation values for the Californian climates (Table 4.1) and due to the fact that the Californian buildings do not have a set back of the windows, the total primary energy use increases with values above 400 W/m^2 more than with the European climates. With the climates of Red Bluff and San Francisco, it even exceeds 100 %, which means that the increased solar gain, when the shading trigger value is 800 W/m^2 or even more, requires more primary energy than the heat gain due to electrical lighting, when the blinds are closed and thus, only a little daylight is entering the spaces.

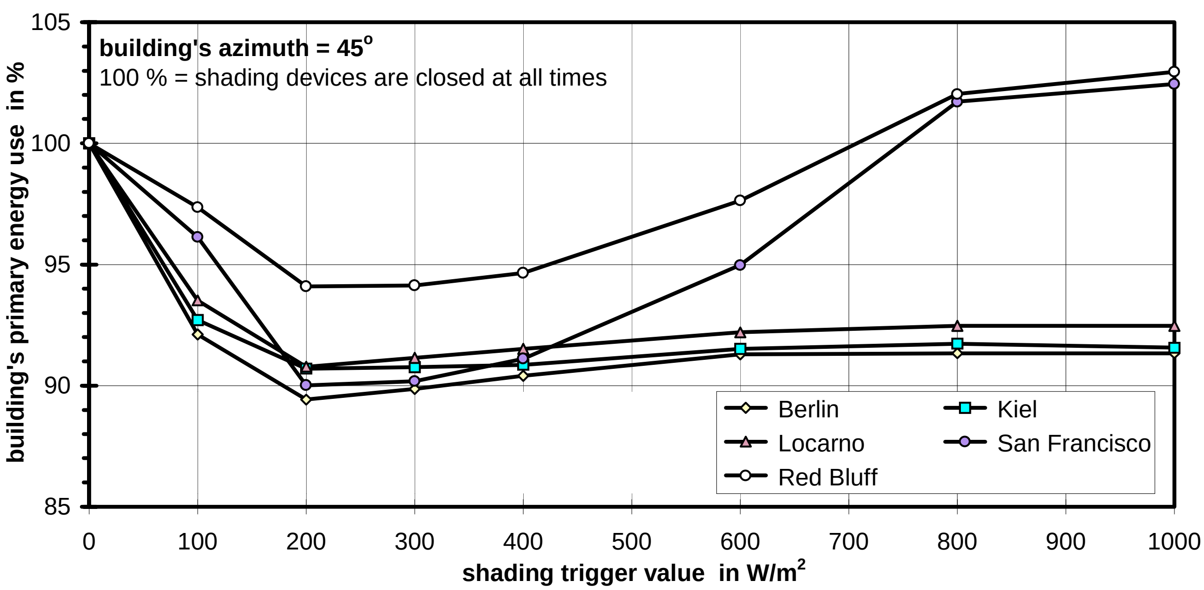

Figure 8.4 : Building’s annual primary energy use in % (100 % = shading devices are closed at all times) versus the shading trigger value (provided light level : 350 lx).

Figure 8.5 presents the dependence of the combined annual primary energy consumption for lighting and cooling (incl. fan energy) on the shading trigger value. Again, the total primary energy consumption for the building (Figure 8.4) with a shading trigger value of 0 W/m^2 represents 100 %. The combined primary energy use due to lighting and cooling (chiller and fans) shows for all climates the minima at 200 W/m^2. Lower shading trigger values (0 and 100 W/m^2) require an increased use of lighting, which demands a higher amount of energy than the saving due to reduced cooling demand. Higher shading trigger values (> 400 W/m^2) cannot decrease the energy use due to lighting furthermore, as the required illuminance is already provided by daylight with a shading trigger value of about 200 W/m^2. But the cooling energy use increases with the shading trigger value, so that these sensor adjustments are unfavorable (see Figure 8.6). However, it becomes clear that it is less critical for the energy efficiency to use slightly higher shading trigger values rather than keeping the blinds more often closed when the solar radiation is below 200 W/m^2.

It is interesting that the range between 200 and 300 W/m^2 always, i.e., for all climates under study, represents the best adjustment of the shading control for the investigated building (Figure 8.5), although the climates chosen are very different (see also Figure 4.1 through Figure 4.5, Chapter 4). As a consequence, all the following simulations were performed using the optimized shading trigger value of 300 W/m^2.

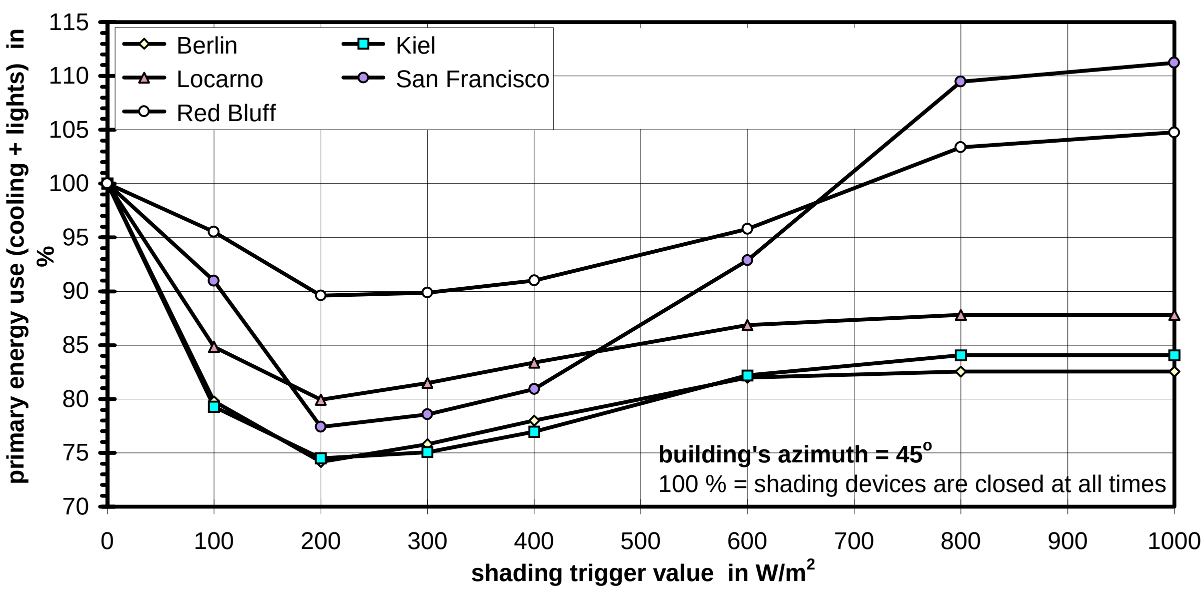

Figure 8.5 : Annual primary energy use of cooling (chiller and fans) and the lights in % of the total energy use versus the shading trigger value (provided light level : 350 lx)

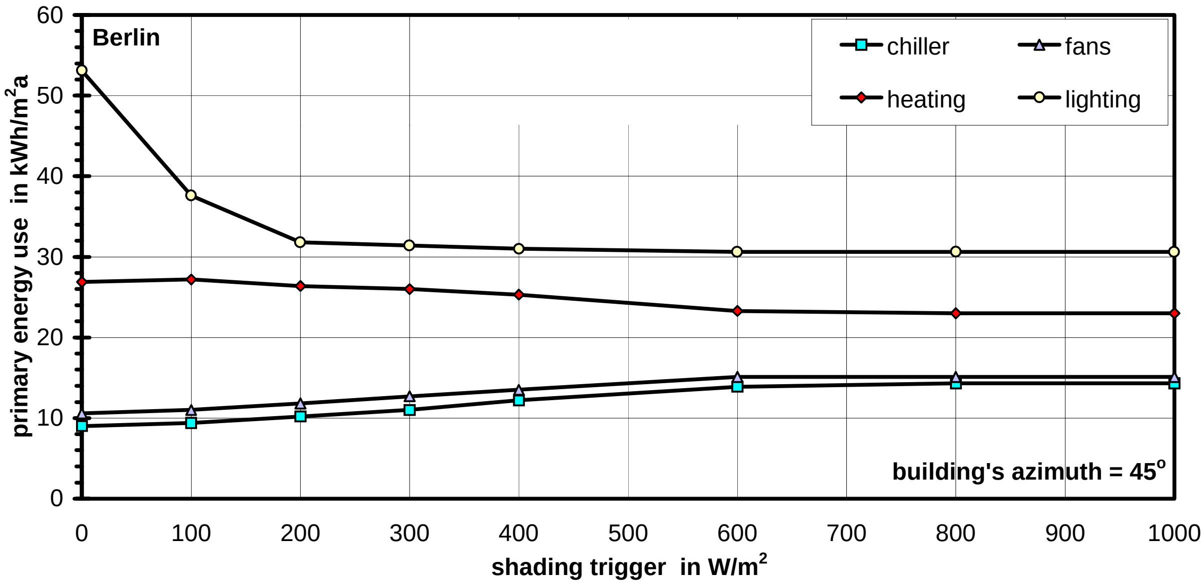

Figure 8.6 : Annual primary energy use of the major HVAC categories versus the shading trigger value (provided light level : 350 lx)

Table 8.1 summarizes the impact of the optimized shading control on the peak cooling load and the annual primary energy use for lighting and cooling (chiller and fans). The basis for this comparison here is the primary energy consumption of the lights, the chiller and the fans without any shading devices operated (unlike Figure 8.4 and Figure 8.5 !). It has to be mentioned that these results (especially Figure 8.6) might not be taken for future comparisons, as the modeled VAV system is not sufficiently adjusted yet.

Table 8.1: Optimized shading trigger values and their impact on the peak cooling load and the annual primary energy consumption for lighting and cooling (chiller + fans)

		reduction of [%]*)	
building location	**optimized shading trigger in W/m²**	**the peak cooling load**)**	**the annual primary energy consumption ***)**
Berlin	300	10 / 27	24
Kiel	300	16 / 30	25
Locarno	300	7 / 24	19
Red Bluff	300	11 / 24	10
San Francisco	300	25 / 32	21

*) compared to the values without any shading
**) entire floor/corner office (highest thermal load)
***) considers **only** the energy use for lighting and cooling (chiller + fans)

8.4 Optimization of the reference system

The size and operation-time of the fans, chillers and boilers, as well as the setpoint temperatures for cooling or heating, offer saving potentials, which are shown in the following paragraphs.

DOE-2E provides the opportunity to size the boilers, the baseboards, fans and chillers automatically. For this purpose, results from the LOADS part are used. But LOADS does not consider, e.g., a floating indoor air temperature, and sometimes inappropriate sizes of fans, chillers or baseboards result from this circumstance. The boiler sizes automatically determined by DOE-2E are appropriate and can be used without problems. To increase the efficiency of the chillers and to reduce their electrical energy use under part load conditions, two chillers (33 % and 66 % of the cooling capacity) are used for this investigation. As DOE-2E can only size one single chiller automatically, the two chillers have to be sized manually.

According to [15] the permissible humidity[18] range in a room should be above 30 % and below 65 % relative humidity and below 11.5 g/kg absolute humidity in order to provide good thermal comfort. DOE-2E offers the opportunity to limit the maximum and minimum of the relative humidity, which are set for all following variations to 30 % and 60%, respectively. Providing this humidity range requires usually humidifying and reheating in wintertime and depending on the climate dehumidifying and reheating in summertime.

With DOE-2E, the best way to size the components is by iteration. After a run using the self-sizing option, the size of the fans, baseboards and chillers can be manually fixed, by considering the data from the SYSTEMS part (different summaries and hourly reports). This procedure is somewhat bothersome, but it leads to properly sized components, which is necessary for the most energy efficient operation. As the hours of operation for fans and chillers influence the cooling loads and the size of the components, the actual energy consumption for each variation can only be found by using an iterative designing process. However, using the data from a Test-Reference-Year for sizing the components does not necessarily lead to realistic sizes of those, as chillers, boilers etc. are usually designed according to the design conditions, e.g., lowest or highest outside air temperature. The Test-Reference-Year might not include these design conditions, resulting in smaller sizes of the components, better part load performance and lower energy consumption.

On the one hand, an optimized “reference” system is supposed to have the lowest possible cooling energy consumption, but on the other hand, the best indoor air quality needs to be provided. Both tasks can not always be met at the same time, because high outside air rates provide best indoor air quality, but in hot and/or humid climates additional energy consumption has to be expected, when

[18] DOE-2E calculates the moisture content only for the central return air, the supply air and the mixed air. Different humidity levels in the particular rooms can not be figured out.

providing high air exchange rates. For these cases, the controlled use of recirculation air can be useful.

In the United States, most modern HVAC systems are equipped with an economizer, which controls the outside airflow rate. An economizer provides 100 % outside air when the outside air is cooler than the return air and reduces the outside airflow to its hygienical minimum, when the dry-bulb temperature (or the enthalpy, depending on the intended control strategy) of the outside air is higher than the corresponding value of the return air. When compared with a fixed minimum outside air rate, the advantage of an economizer is the reduced cooling energy use, when a higher outside air rate (with cooling capacity) is supplied to the space. This advantage can be used, if the cooling load is mostly caused by internal heat sources, or during the early morning hours and in the transient seasons. The recirculation airflow decrease the cooling energy use, but pollutants, e.g., VOCs[19], CO_2, odors, and other unwanted aerosols are not always removed completely by the filter units (bad maintenance), so that the indoor air quality can be worse than by supplying only outside air.

Modern VAV systems in European office buildings usually operate only with outside air and therefore avoid a recirculation of return air, which could favor the sick building syndrome [21]. Probably higher energy consumption is the result, but in most cases, using an optimized heat recovery unit can minimize additional energy use. Beside others, this issue is investigated in the following section.

8.4.1 Optimization of the schedules

In order to reduce the energy consumption, the building should not be completely cooled, vented or heated on weekends and during the night hours. Extended operation hours in the morning need not generally cause a higher annual energy use, as longer running hours can decrease the cooling loads, especially in the first running hours of a day. This schedule could result in smaller sizes of chillers and fans. These smaller components may have smaller or larger energy consumption, depending on the operation hours, the peak loads and part load ratios. In order to determine the energy consumption, the sizes of chillers and fans have to be adjusted to the different schedules.

[19] VOC : Volatile Organic Compounds (e.g.: acetone, toluene, ethanol)

Assuming that the use of an economizer is the most energy-efficient way of air-conditioning a building, the following comparisons have been made. First, the following operation variations are investigated with several running hours for fans and chillers :

a) The same setpoint temperatures[20] and identical operation hours for fans, cooling and heating are used for weekends *and* weekdays.

b) The heating setpoint temperature is reduced on weekends.
Fans and chillers are switched off on weekends (no cooling).
Fans are operated during 8^{oo} - 17^{oo} in wintertime on weekdays.

c) The heating setpoint temperature is reduced on weekends.
Fans and chillers are switched off on weekends (no cooling).
Fans are operated during 8^{oo} - 17^{oo} in wintertime on weekdays.
An economizer is being used for reducing the cooling use.

d) Setpoints are reduced on weekends.
Fans and chillers are switched off on weekends.
Fans are running during 8^{oo} - 17^{oo} in wintertime on weekdays
100 % outside air is supplied
(this variation is made just for the most advantageous fan running hours of the economizer mode, i.e., **c)**.)

Secondly, the energy use and the demand of the variation a) to c) for different time schedules and all five climates investigated are compared. The primary energy consumption of cooling and ventilation and the required cooling capacities can be used for determining the most advantageous time schedule for fans and chillers. After having found the most energy-efficient economizer operation schedule, this will be compared with a VAV system, which always uses 100 % outside air. Finally, the best solution for each of the climates is presented, by showing the particular portions of the building's primary energy consumption.

Exemplary for Red Bluff[21], Figure 8.7 presents the annual energy use for the variation a) to c) with different fan operation hours. It is not surprising that operating the plant just five days a week, instead of seven days, is more energy efficient. However, it is hard to tell whether the economizer offers additional savings and which categories of use are most advantageous.

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[20] compare **Error! Reference source not found.**

[21] Figures for the other climates can be found in the appendix (Figures A10 to A17).

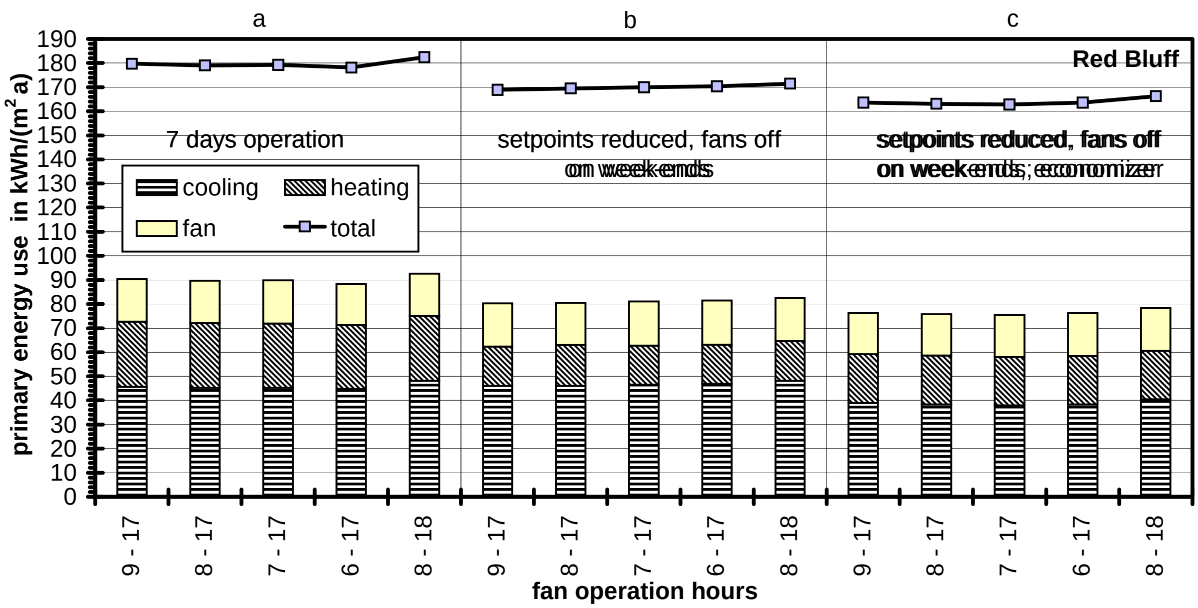

Figure 8.7 : Building's annual primary energy use for Red Bluff in kWh/(m^2 a) versus the operating hours of fans and chillers.

To distinguish the particular energy use of each of the categories, the energy saving potentials of the operation variations b) and c) related to variation a) for five different operation hours for fans and chillers are shown in Figure 8.8.

It can be seen that the different hours of fan operation result in different cooling energy savings. Operating the plant five days a week without an economizer demands more cooling energy than the 7-day operation. The 5-day operation, however, causes higher cooling loads, which leads to higher cooling energy consumption, although the running time for the chillers is shorter. The economizer saves a lot of cooling energy, but simultaneously provides less heating energy savings. However, this disadvantageous characteristic of an economizer only occurred with the Red Bluff climate. The economizer allows 100 % outside air to enter the system in the early morning hours. This very dry outside air requires more humidification and reheating than a mixture of recirculation and outside air to meet the humidity requirements, so the heating energy consumption increases in Red Bluff when using an economizer instead of a fixed recirculation air ratio.

The total primary energy savings reflect that the cooling energy savings due to the economizer use are bigger than the reheating energy consumption for the humidification. For Red Bluff, total primary energy savings with the economizer of more than 9 % can be achieved (Figure 8.8) when compared with the 7-day operation mode without economizer.

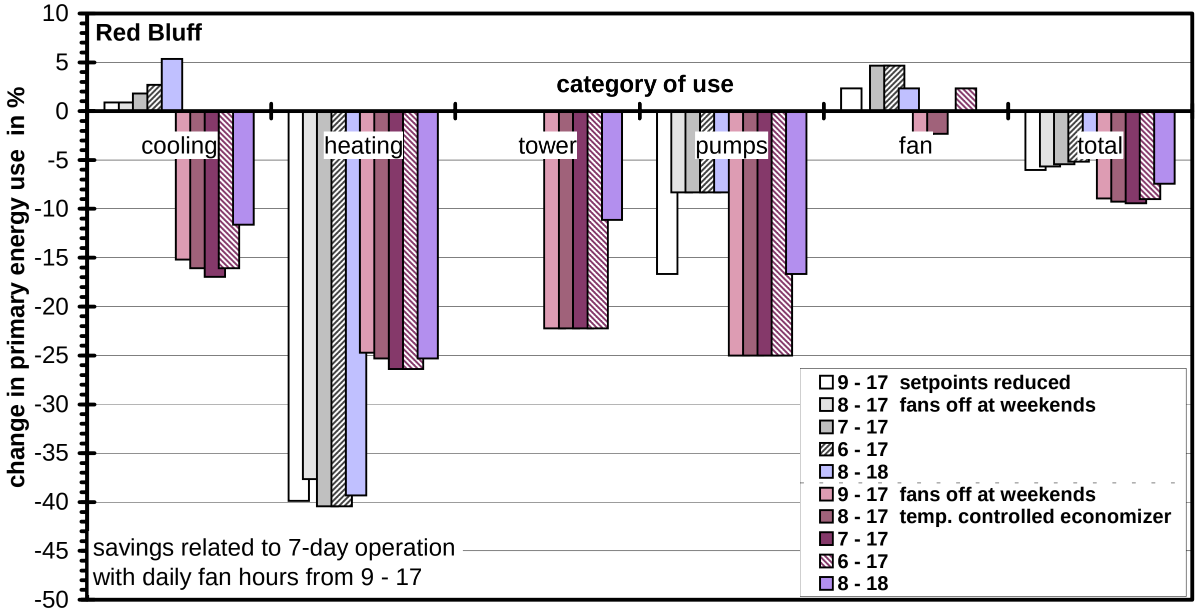

Figure 8.8 : Primary energy savings in % due to different operation strategies and fan running hours for Red Bluff.

Figure 8.9 to Figure 8.11 present the building's primary energy consumption (Figure 8.9), the primary energy use for cooling and fans (Figure 8.10) and the building's cooling load (Figure 8.11) for the five climates investigated. To show the most energy efficient fan schedule, each of these Figures compares the operation strategies a) to c) and five different fan operation hours.

For all climates investigated, using an economizer can reduced the primary energy consumption (Figure 8.9). Although no big differences in primary energy use can be noticed within the economizer mode, it becomes clear that for all climates it is more energy efficient to start the system before the people enter the building and stop immediately after the occupation hours rather than running the fans and chillers after the office hours. As the differences of the primary energy use are small, other parameters have to be investigated to determine the best fan schedule.

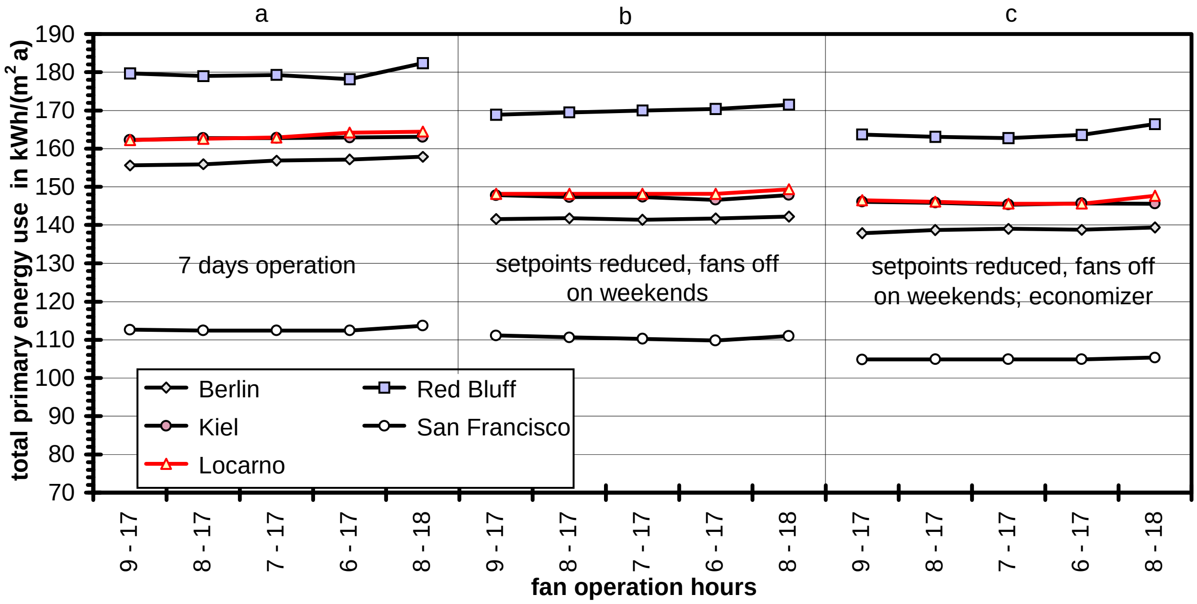

Figure 8.9 : Building's annual primary energy use in kWh/(m^2 a) versus the operating hours of fans and chillers for the five climates investigated.

Figure 8.10 shows the primary energy consumption due to cooling and ventilating (fans) for the different fan schedules, climates and operation strategies. The influence of the different fan hours on the energy use of the fans and chillers is very small. For each particular climate the fans are using almost the same amount of energy, because with longer operation hours the building can be cooled with less air (smaller fans). This compensates for the longer fan running time.

Figure 8.11 shows the building's cooling load determining the required cooling capacity of the chillers for cooling and dehumidifying. It becomes clear that starting fan operation earlier than the arrival of the occupants reduces the cooling load remarkably.

Considering the results shown in Figure 8.7 to Figure 8.11, the fan operation schedule "7 - 17" seems to be the most energy efficient schedule. The fans and chillers are switched on 2 hours before the office hours and are stopped immediately after the occupants have left the building. This schedule is used for all further investigations.

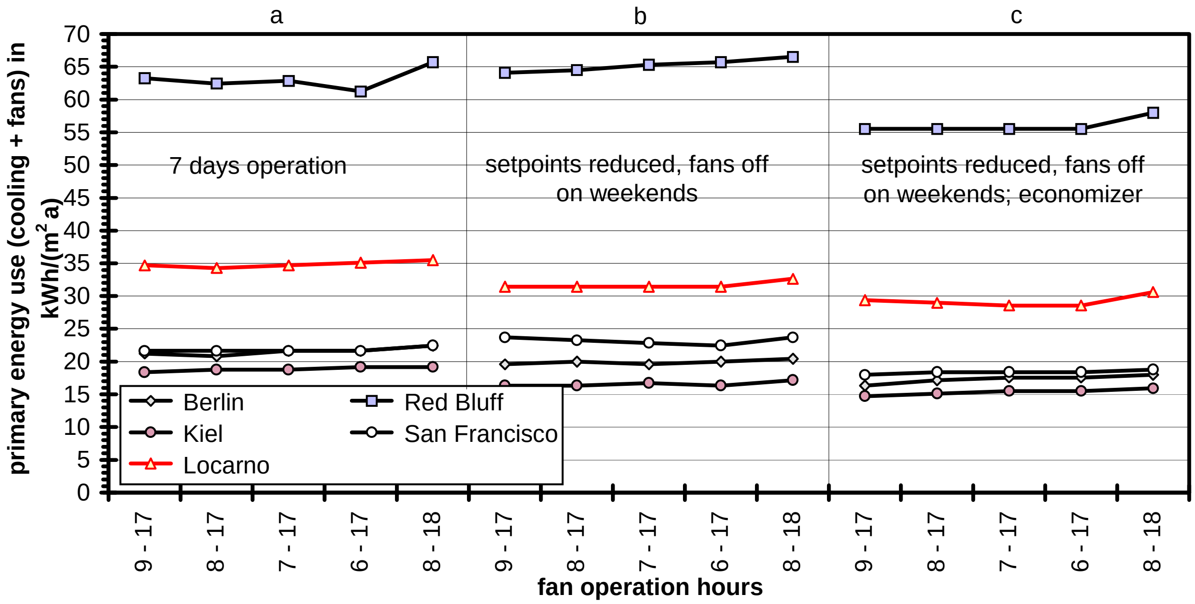

Figure 8.10 : Annual primary energy use for fans and chillers in kWh/(m^2 a) for all five different climates versus the operating hours of fans and chillers.

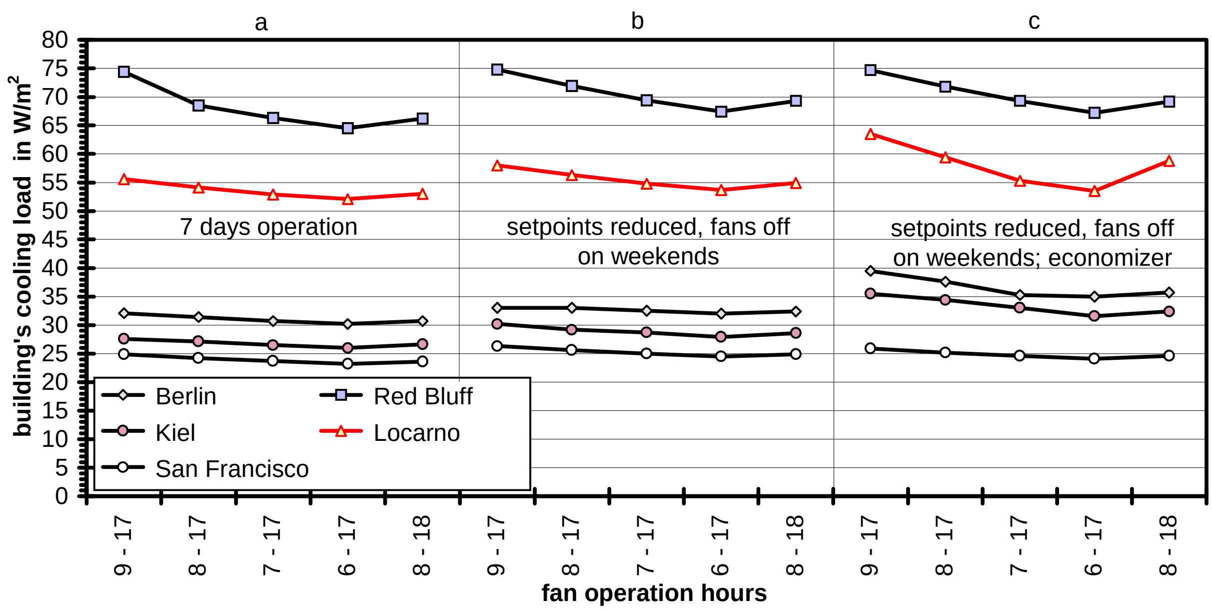

Figure 8.11 : Building's cooling load (average of all spaces) for all five climates and different strategies as a function of the operating hours of fans and chillers.

The comparison between a VAV system equipped with an economizer and a system using 100 % outside air is presented for all five climates in Figure 8.12. For the German climates investigated there are almost no differences, but for Locarno, San Francisco, and especially Red Bluff, the building's primary energy consumption increases when supplying 100 % outside air instead of using an economizer (up to 12 % in Red Bluff).

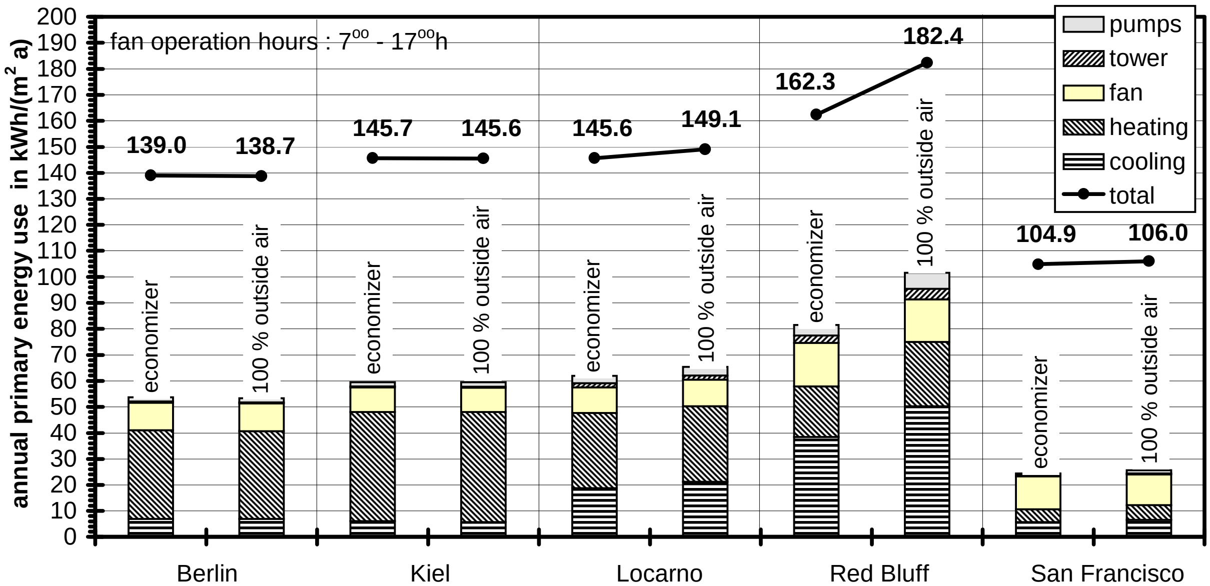

Figure 8.12 : Comparison of primary energy use with economizer mode and using 100 % outside air for all five climates investigated and the fan operation hours "7 - 17"

8.4.2 Final sizing and energy use of the reference system

According to Figure 8.12, the most energy efficient reference VAV system can be chosen for each of the climates. For the German climates, the VAV system with 100 % outside air is being used. The other climates get a VAV system with temperature-controlled economizer. These reference systems are being used as basecase to evaluate the saving potentials of the alternative cooling strategies in the following chapters.

The most important energy use components for the reference VAV systems and all climates investigated are shown in Figure 8.13 Table 8.2 presents the shares for each category to show the differences between the chosen climates. In this case, the total energy consumption for the investigated floor represents 100 %.

The components of the cooling energy consumption show the saving potential possible with other cooling strategies. Not only the cooling energy, but the energy for the cooling tower, the fans and heating might be involved as well. Having the climatic conditions and the characteristics of the alternatives in mind, the potential to provide thermal comfort and save primary energy can be roughly predicted, or estimated, respectively.

The two German climates require a lot of heating (25 - 30 %), but only a very little cooling (ca. 5 %). For these climates, expensive, i.e., in first and operation costs, alternatives to compressive cooling have almost no chance to be profitable. The very moderate climate of San Francisco neither requires a lot of cooling nor a lot of heating and the primary energy use of the lights and the equipment makes up about 75 % of the entire primary energy consumption. However, considering the very temperate climate in San Francisco, alternative cooling strategies seem to be designated for this region.

Locarno, and especially Red Bluff, offer better conditions for alternative cooling strategies in terms of pay-back time, as the cooling energy consumption makes up a higher share. Best potentials for alternative cooling strategies can be expected for Red Bluff, where no dehumidification is needed. The humid climate in Locarno limits most alternative strategies, because rather low coolant temperatures are frequently required in order to dehumidify the outside air sufficiently. Absorption chillers can replace their compressor-driven counterparts completely, however, the primary energy use might be higher. Just desiccant cooling does not use cooling below the dewpoint to dehumidify the moist outside air and therefore, this seems to be a competitive alternative in Locarno.

The primary energy consumption shown represents a very energy efficient building, equipped with a modern, well-adjusted HVAC system, which becomes clear when comparing with the energy use of typical office buildings. It has to be considered that both the internal and external cooling loads and the performance and operation schedule of the HVAC system has been optimized. The achieved *primary* energy intensity of the floor with the reference VAV system varies between

105 and 160 kWh/(m^2 a), whereas a *primary* energy intensity of about 500 - 700 kWh/(m^2 a) can be assumed for existing office buildings in the United States and about 200 -700 kWh/(m^2 a) for existing office buildings in Central-Europe (compare Chapter 5) [21,55].

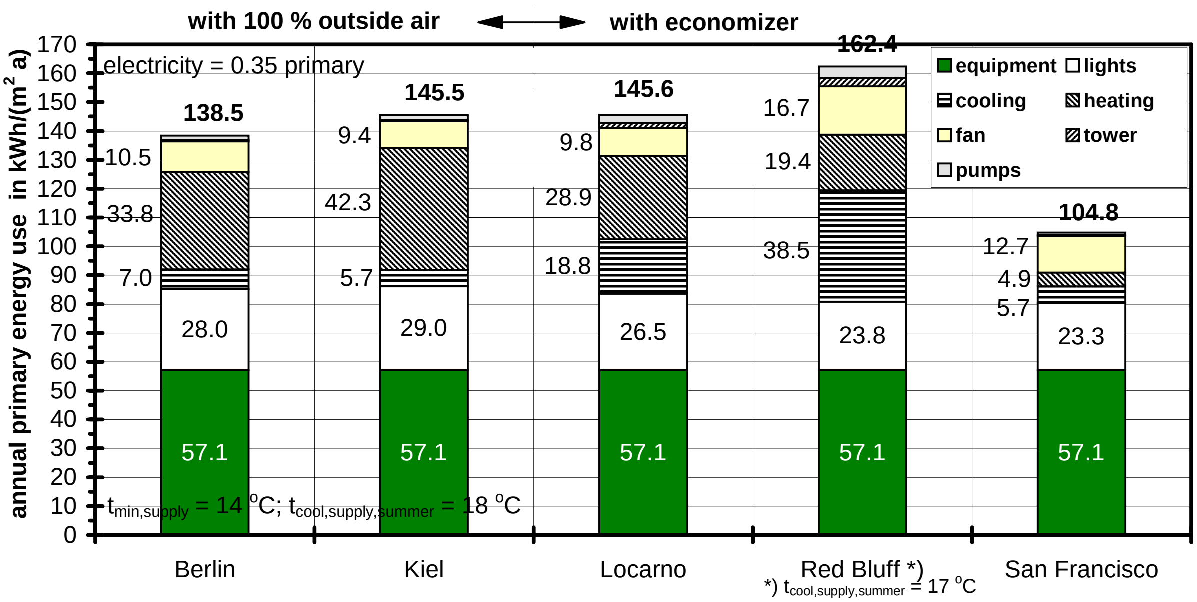

Figure 8.13 : Reference energy consumption of the building for the five different climates investigated, considering a comfortable indoor air humidity range.

Table 8.2 : Components of the primary energy consumption for the five different climates investigated (building's primary energy use = 100 %)

	portion of the annual primary energy consumption in %						
location	**equip-ment**	**lights**	**cooling**	**tower**	**heating**	**pumps**	**fan**
Berlin	41	20	5	< 1	24	1	8
Kiel	39	20	4	< 1	29	1	7
Locarno	39	18	13	1	20	2	7
Red Bluff	35	15	24	2	12	2	10
San Francisco	55	22	6	< 1	5	1	12

Table 8.3 presents the specific (related to the floor area) sensible peak cooling loads and required cooling and heating capacities for the five different climates investigated. The building's average and the highest cooling load are shown. Additionally the required total cooling capacities (sensible and latent) and the heating capacities for the supply air and the baseboards are presented.

Table 8.3 : Specific thermal loads of the investigated floor of the building for five different climates

	thermal peak loads W/m^2			
			required capacity for	
	sensible cooling load *)		cooling**)	heating
location	**maximum**	**building average**	**(sensible + latent)**	**supply air + baseboards**
Berlin	34	21	38	37
Kiel	35	21	32	41
Locarno	36	23	59	41
Red Bluff	65	35	70	42
San Francisco	48	26	25	23

*) The sensible cooling loads are given for a constant indoor air temperature of 23 ^{o}C, because DOE-2E provides cooling loads of particular rooms only in the LOADS part, which considers fixed indoor air temperatures. As this temperature is allowed to drift during the day, the resulting 'real' loads might be different.

**) The SYSTEMS part calculates the systems cooling load (including latent load and load due to dehumidifying the supply air) considering the floating indoor temperature, but only for the entire system (here : entire floor), and not for each particular room.

Table 8.4 summarizes the characteristic design data for the five climates investigated.

Table 8.4: Design data for the components of the reference system (one floor = 700 m^2) for the five different climates

		location				
variable	**unit**	**Berlin**	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
$\dot{Q}_{heater}$	**kW**	26	29	28	29	16
$\dot{Q}_{chiller}$	**kW**	9/18[*)]	8/16	13/26	16/32	6/12
$\dot{V}_{supply,max}$	**m^3/h**	4700	4300	7400	11500	5700
$t_{supply,min}$	**°C**	14/18[**)]	14/18	14/18	14/17	14/18
$t_{supply,max}$	**°C**	21	21	21	21	21

*) : two chillers are used to avoid ineffective part load operation

**) : the lower temperature is being used in case of dehumidification is needed, the second is the supply air temperature during the cooling season

8.5 Influence of constant indoor air setpoint temperature

To demonstrate the benefit of a floating indoor air setpoint temperature the primary energy consumption of the reference VAV systems were also calculated with a fixed setpoint temperature. For these calculations the indoor air temperature was set to 24 °C. The required supply airflow and cooling capacity were adjusted accordingly.

Figure 8.14 presents the increased primary energy use of the VAV system with fixed indoor air temperature when compared with the floating setpoint. Additionally, the increase of the cooling peak load is shown in the last category of the ordinate. The particular value for each category with the reference system represents the base for the comparison. Fixing the indoor air temperature to 24 °C requires about 7 to 19 % more primary energy for the HVAC system. The total primary energy consumption of the building (incl. lights and equipment) will be increased by 2 to 10 % and the peak cooling load by 17 - 43 % depending on the climate.

Figure 8.15 shows the components of the increased primary energy consumption of the HVAC system in detail. In this Figure, the difference in system's primary energy use with constant and floating setpoint temperature represents 100 % for each of the climates. The single categories of the HVAC system presented illustrate their contribution to the increased energy use of the HVAC system. As expected, the excess energy for cooling and the fans are the major influences.

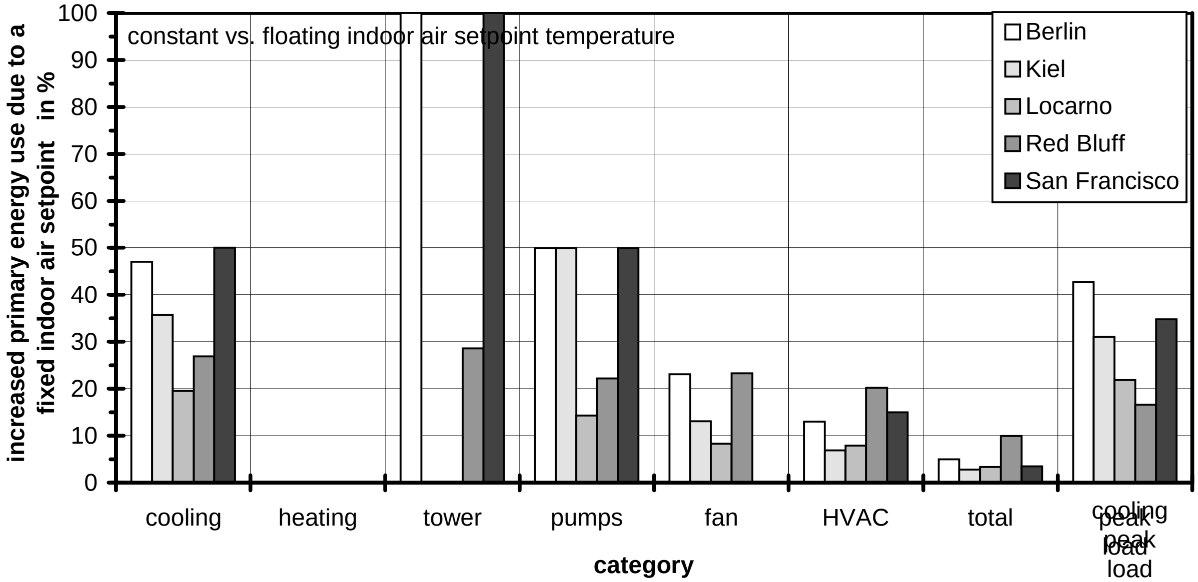

Figure 8.14 : Increased building's primary energy use due to a fixed indoor air setpoint temperature for the five climates investigated.

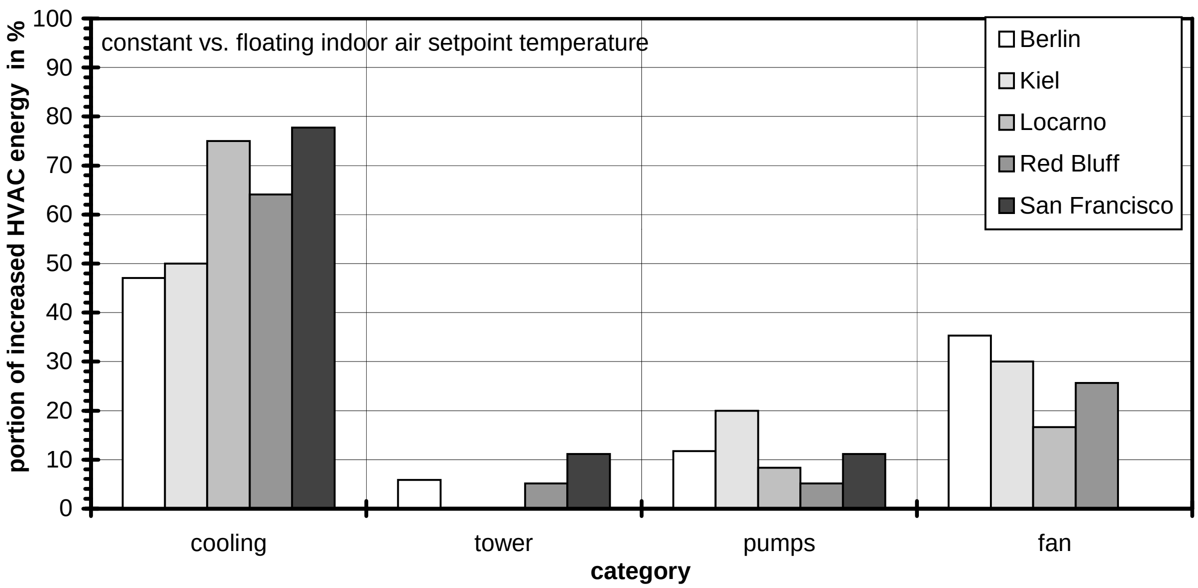

Figure 8.15 : Portions of increased primary energy consumption of the reference VAV system due to a fixed indoor air setpoint temperature for the five climates investigated by category (refers to category "HVAC" in Figure 8.14).

8.6 Air-cooled condenser instead of cooling tower

The heat removed from the spaces is commonly been rejected to the environment by utilizing a cooling tower. A cooling tower evaporates water into an air stream to cool the circulating cooling water from the condenser of the chiller. To prevent increasing concentrations of dissolved solids (scale control) and other impurities, the cooling water might be treated and a certain amount of the circulating water needs to be changed continuously [3]. Thus, not only the water being evaporated to reject the heat is consumed. A cooling tower consumes about 6 kg/h of water per kW reject heat [34] which can add up to a significant annual amount to consider when assessing the environmental impact of a cooling strategy.

The hourly water consumption $\dot{m}_{water}$ of a cooling tower can be determined more accurately by using the following equation [34] :

$$\dot{m}_{water} = \frac{\dot{Q}_{reject}}{2500} \cdot \left(1 + \frac{1}{E-1}\right) + \dot{m}_{spray} \quad \text{in kg/s} \tag{1}$$

with : $\dot{Q}_{reject}$ = $\dot{Q}_{cool} + \dot{Q}_{chiller,el}$ = reject heat ≈ 1.2...1.4 * $\dot{Q}_{cool}$ depending on the COP of the chiller in kW

$\dot{Q}_{cool}$ = heat removed from the spaces

$\dot{Q}_{chiller,el}$ = electrical power demand of the chiller

2500 = evaporation heat in kJ/kg

E = d_{max}/d

d_{max} = permissible water hardness in ^{o}dH
(1^{o} dH = 0.18 mol $CaCO_3$ /m^3 ; usual values ≈ 20 - 25 ^{o}dH)

d = actual water hardness in ^{o}dH

$\dot{m}_{spray}$ = water losses

Unfortunately, DOE-2E does not report the water consumption of the cooling tower. Therefore, the water consumed is determined by using Equation 1 and the reported simulation results for the heat removed from the spaces and the energy input of the chillers. Table 8.5 presents the water consumption of the cooling towers for the five different climates investigated. The ratio of the permissible to the actual water hardness is set to 1.6. Water losses of 1 kg/kWh are considered.

Table 8.5 : Annual water consumption of the cooling tower according to Equation 1 considering spray losses of 1 kg/kWh reject heat.

	location				
parameter	**Berlin**	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
heat removed from the spaces [MWh/a]	4518	3810	12329	26080	3717
heat rejected in the cooling tower [MWh/a]	6224	5250	16942	35515	5150
water consumed [kg/a]	30125	25410	82000	171895	24925

Instead of using a cooling tower, the heat can be rejected from the condenser by air. Such an air-cooled condenser demands normally slightly more electric energy for the fans but no treated water is been consumed.

The reference VAV system with a cooling tower is compared with a VAV system with an air-cooled condenser by evaluating the annual primary energy consumption. The results are presented in Figure 8.16. Only the hot and dry climate of Red Bluff leads to significantly increased energy consumption when an air-cooled condenser is been used for rejecting the heat removed from the building. In this case the HVAC system with an air-cooled condenser demands about 5 % more primary energy than the one with a cooling tower. The other four climates do not show remarkable differences in primary energy use.

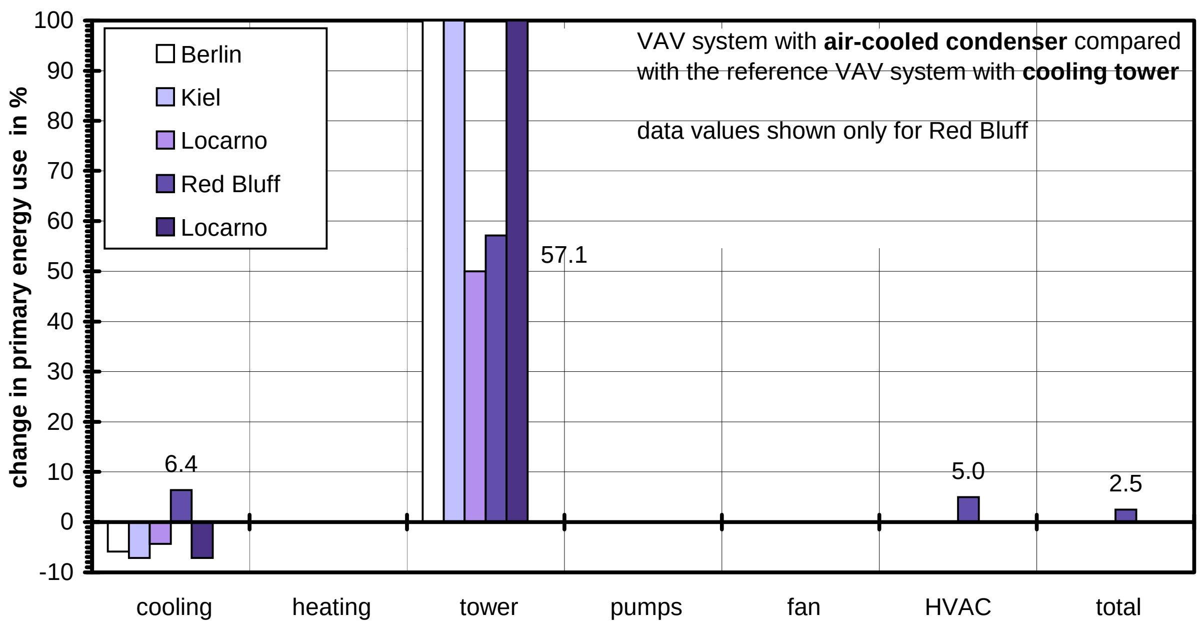

Figure 8.16 : Changes in primary energy consumption when the use of a cooling tower is compared with an air-cooled condenser

9. Night Ventilation

Night ventilation takes advantage of the cooling potential of the outside air at night and the thermal storage capacity of the building. If the outdoor air temperature is lower than the indoor air temperature and the radiant temperature, a supplied outside airflow is cooling the building's surfaces and the room air. A sufficient thermal storage capacity is required to cool the spaces during the daytime by storing the heat released, when the masses can be discharged at night. Night ventilation can be realized by venting the building mechanically with fans or by using the exterior windows, interior doors, the wind and pressure differences between outside and inside (natural ventilation).

Night ventilation can be an alternative to compressive cooling [35,36], but not in general. However, this strategy can be used to support the conventional compressive chiller by pre-cooling the building's structure at night. A pre-cooled building buffers the cooling peak load, especially when the peak occurs in the first hours of occupancy. By allowing the inside air temperature to float during the day up to the designed threshold limit [4,15], the cooling energy demand in the afternoon can be reduced as well as the daily cooling energy consumption. With additional night ventilation a smaller chiller size and shorter running times of the chiller can be achieved reducing the electrical energy use of the chiller. Possibly the size of the fan (or the highest volume flow) can be cut down as well, but due to longer diurnal operation hours of the fan (night and day), the electrical energy use might not change significantly.

Table 9.1: Night ventilation : Percentage of night hours in summer below a certain dry-bulb temperature for the five climates

dry-bulb temperature in ^{o}C	location				
	Berlin	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
10	26	22	16	3	4
15	69	69	50	23	88
20	97	99	91	58	99
25	100	100	100	88	100
30	-	-	-	98	-
35	-	-	-	100	-
$\bar{t}_{dry,night}$	12.6	12.3	14.3	18.8	12.7

Example : 50 % of the night hours during summer in Locarno are below 15 ^{o}C

Table 9.1 shows the percentage of night hours (from 22^{oo} to 8^{oo} h) for the five climates chosen during the summer (from May to October) being below the corresponding dry-bulb temperature. Figure 9.1 presents the dry-bulb temperature, the dewpoint temperature and the global solar radiation for the five climates during one characteristic cooling peak day.

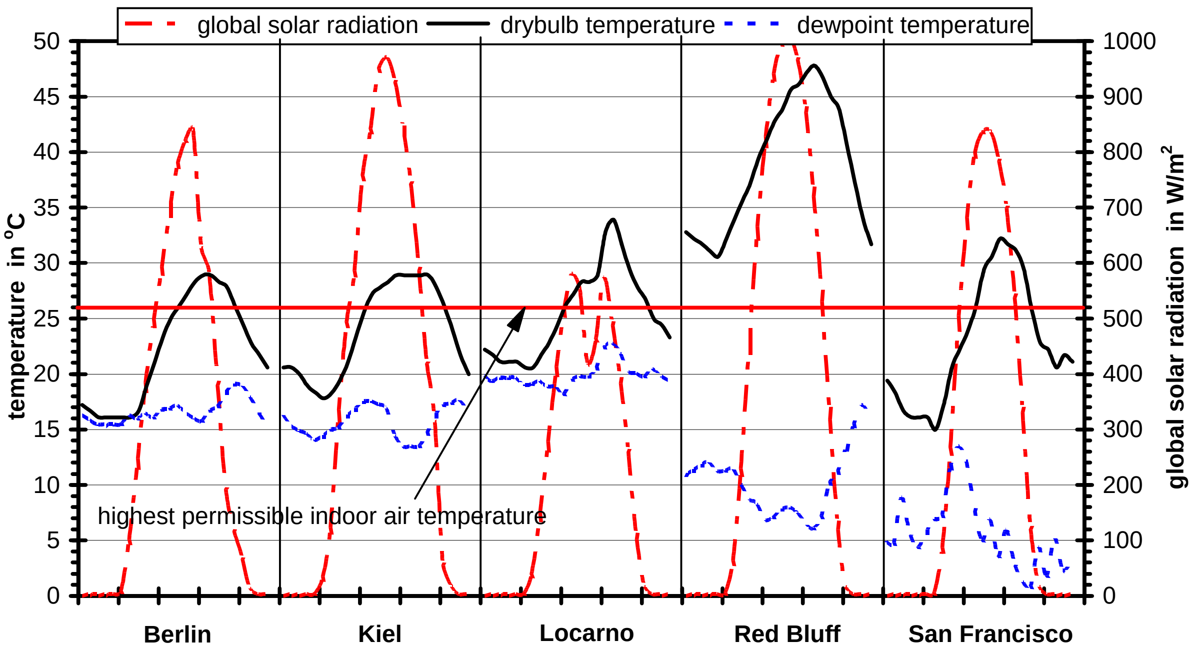

Figure 9.1 : Hourly outside temperatures and global solar radiation during one particular cooling peak day for each of the five climates investigated.

As the weather conditions in summertime in Berlin and Kiel do not differ very much from each other (Table 9.1, Figure 9.1), night ventilation is investigated only for Berlin, representing the climate of the major German cities. The rather high humidity level in Locarno has to be considered apart from the dry-bulb temperatures, which will be done in the following paragraphs. Red Bluff offers unfavorable boundary conditions for night ventilation, because the outside air (dry-bulb) temperature at night is often even above the highest permissible indoor air temperature (42 % of the nighttime in summer are warmer than 20 °C and 12 % are warmer than 25 °C; compare Table 9.1). However, there are even in Red Bluff some days with outside air temperatures at night, which provide a certain cooling potential. A preliminary study showed that is not possible to provide thermal comfort in the building's spaces only by means of night ventilation, i.e., no chiller is been used additionally.

Considering the weather data presented above, only the climates Berlin, Locarno, and San Francisco will be investigated thoroughly with using all night ventilation strategies. Some results for Red Bluff (preliminary study) are presented utilizing a VAV system with compression chiller and mechanical night ventilation in addition.

9.1 Natural night ventilation

If the windows of a building are to be used for venting and cooling the rooms with outside air, special, sophisticated window elements (small and large openings with control devices) should be installed in order to make individual air exchange rates possible. The control devices can direct dampers to close the respective supply air duct or switch off the HVAC system for rooms with opened windows to avoid waste of conditioned air and energy [22]. In buildings without any mechanical ventilation, however, a reliable control of the air exchange rate is quite not possible, as many different parameters (wind velocity and direction, temperature differences, height of the building and the space, window size, interior openings etc.) besides the users themselves have an influence. In areas with high exterior pollution levels (urban areas) natural ventilation cannot always be recommended for venting and cooling the building either, although the major air exchange is made during the night hours. As there is no way to control the indoor air humidity, uncomfortable thermal conditions are very likely to occur both in regions with high absolute humidity levels in summer, e.g., Locarno, or low absolute humidity levels in winter, e.g., Berlin.

Unfortunately, another backside of natural night ventilation has to be considered. For security reasons, windows might not be left open during the night. However, this issue is not been considered in this study.

For this investigation, three different air exchange rates (infiltration via windows) are taken into account, so that the potential of natural night ventilation can be assessed. Figure 9.2 shows the outside air exchange rates assumed, which are supposed to represent the user-operated natural ventilation by opening and closing the windows and doors manually.

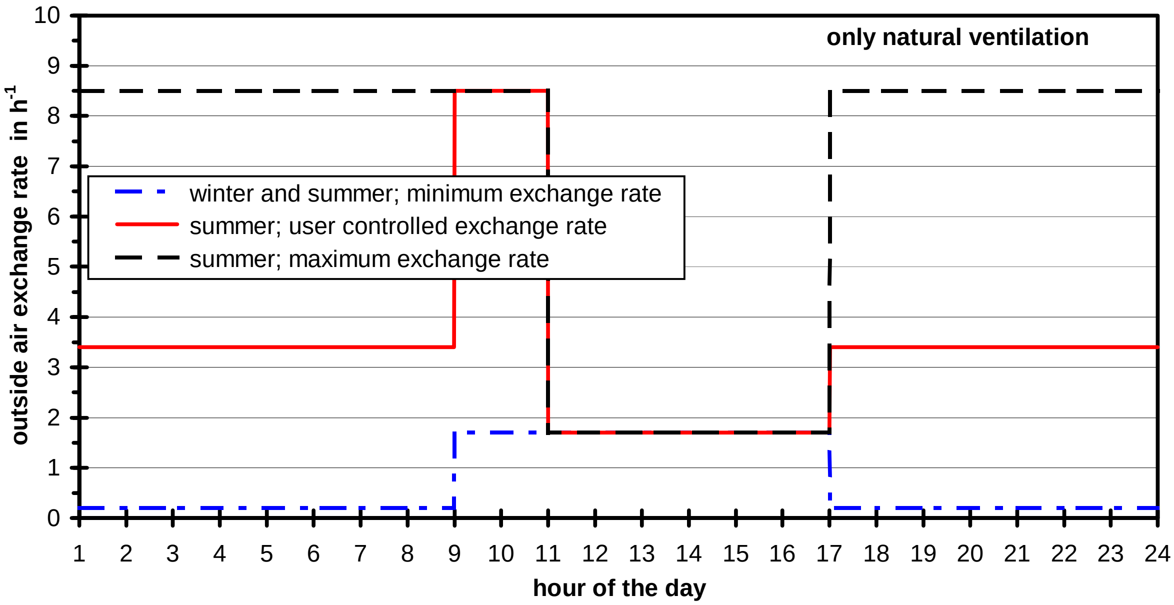

Figure 9.2: Outside air infiltration rates for the building with natural night ventilation

Explanations to Figure 9.2:

The *'minimum exchange rate'* provides just the outdoor air requirement [5,15] during the occupation hours for the whole year. At night only infiltration as a result of leaky facades is considered. This minimum air exchange rate and the supply airflow for the heating season provided by the reference VAV system are identical during the working hours.

The *'user-controlled exchange rate'* is supposed to represent a rather realistic usage of the windows in summertime. Only some windows and interior doors of the investigated floor are left open by chance during the night, but in the first morning hours the occupants open most of the windows, so that the (cool) outside airflow into the building is increased. All windows are closed after the first two hours for the rest of the working day, however, the hygienically required outside air rate is still entering the spaces through temporarily opened windows. Some windows remain opened after the working hours.

The *'maximum exchange rate'* gives an idea of which cooling potential could be available, if the windows would be used in the most effective way. The exterior windows and interior doors are manually opened at the end of the working day and stay opened until the next morning. Comprehensive ventilation of the building at night hours in summertime can be achieved in this case. During the occupation hours just the hygienically necessary outside airflow is provided.

9.1.1 Thermal comfort

Thermal comfort is basically being assessed by checking whether the spaces are cooled or heated sufficiently. The relative humidity (RH) levels inside are controlled as well, considering the comfort limits (30 % $\leq \varphi_{Space} \leq$ 60 % RH). DOE-2E provides yearly summary reports showing the hours of fan operation being within a certain range of return air temperature and humidity ratio ($\cong$ mean indoor air temperature/humidity). Only hours during the occupancy are considered.

<u>Berlin :</u>

Figure 9.3 presents DOE-2-calculated indoor air temperature profiles at the very cooling peak day in Berlin[22]. The three different natural exchange rates are considered and the derived three mean indoor air temperature profiles are compared with the air temperature provided by the reference VAV system (Chapter 8.4.2).

Comfortable indoor air temperatures (**Error! Reference source not found.**) can be achieved in Berlin either with the *'user-controlled'* or the *'maximum'* air exchange rate at the cooling peak day presented. The corresponding DOE-2E summary reports state no hours with unacceptable indoor air temperatures. Comparing the curves presented in Figure 9.3 it becomes clear that a high air exchange rate at night provides a remarkable cooling potential, but the windows should remain closed after the working hours, as the outside air temperature is not low enough to cool the building during these hours (compare the curve for the *'user-controlled'* and the *'maximum'* air exchange rate between 17^{oo}h and 20^{oo}h; Figure 9.3). Opening the windows after, e.g., 22^{oo} h (depending on the temperature difference) would be favorable, but difficult to realize manually.

[22] The temperature profiles for Locarno (Figure A18) and San Francisco (Figure A19) can be found in the appendix.

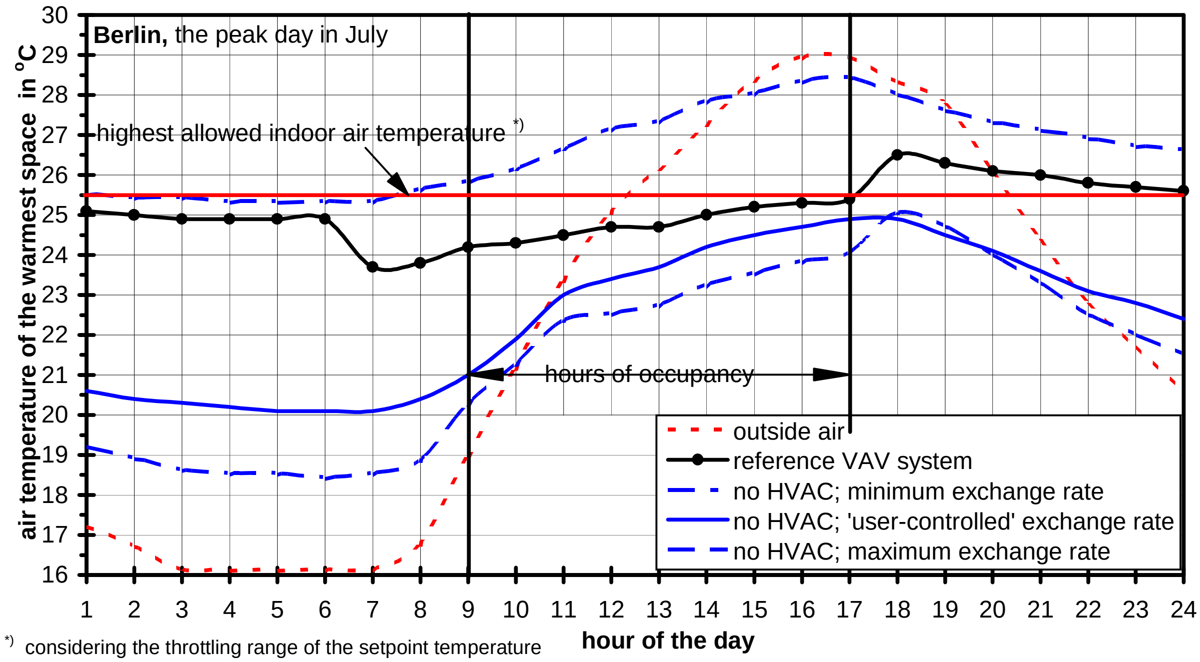

Figure 9.3 : Comparison of indoor air temperatures by using natural ventilation with different air exchange rates and the reference VAV system (Berlin).

Just considering the dry-bulb temperatures, natural night ventilation seems to be appropriate an alternative cooling strategy in a moderate climate like Berlin, but it remains dubious whether a reliable exchange rate can really be supplied to the spaces by using just the windows. Automatically controlled opening devices might be able to provide the required air exchange rate under design conditions and schedules for venting can be accomplished. However, it is still fairly uncertain which air exchange rate would take place when the boundary conditions are changing or if the maintenance of the devices does not happen to be regular.

Unfortunately, thermal comfort cannot always be provided sufficiently considering the climate of Berlin, as the outside air moisture content often is high enough (up to 14 g/kg, compare Table 4.3) to cause comfort problems inside. The relative humidity in the building cannot be controlled, so it exceeds 60 % RH at about 210 h of the year corresponding to about 12 % of the office hours during the cooling season. About 40 h of the year can even be above 70 % RH. Additionally 553 hours in wintertime (≅ 47 % of the heating season) are too dry, which might also be sensed being uncomfortable.

Locarno :

Satisfying thermal comfort can often not be achieved with natural night ventilation for the building located in Locarno (Figure A18; appendix). Both the indoor air temperature and the relative humidity level restrict thermal comfort in summer.

About 150 h (≅ 12 %) of the working hours during the cooling season are registered being too warm. More than 190 h (≅ 11 %) are reported to have a higher relative humidity than permitted. But these relative humidity levels cannot be used for assessing thermal comfort at this point, as it is related to the indoor air temperature, which already is too high during a remarkable number of hours. Assuming indoor air temperatures within the allowed range, many more hours would be above 60 % RH. The inside air absolute humidity in Locarno sometimes increases up to 14-18 g/kg during the working hours, whereas 11.5 g/kg is the upper permissible limit for thermal comfort. Problems with dry air in wintertime are very likely, too. The lower threshold limit for the relative humidity is not provided at about 770 h of the daytime hours, which corresponds to about 66 % of the working time during the heating season.

San Francisco :

Natural night ventilation provides indoor air temperatures within the comfort range for almost the whole cooling season in San Francisco, although the indoor air temperature at the very cooling peak day is way above the threshold limit (shown in Figure A19; appendix). The indoor air temperature of the warmest space happens to be above the permissible value at a few hours of the year considering the *'user-controlled'* exchange rate. This ought to be acceptable, however, the relative indoor air humidity was calculated to be higher than 60 % RH at about 150 h, which is about 8 % of the cooling season. This unfavorable result is due to the lacking temperature control with the natural night ventilation simulated, i.e., sometimes, the indoor air temperature is clearly below the allowed value. Considering room temperatures close to the permissible level, problems with exceeded humidity levels do not occur any more. Nevertheless, during the heating season about 140 h were reported being too dry (≅ 12 %).

The indoor air temperatures and relative humidity levels of the three climates investigated during four days of the cooling season are summarized and presented in the Figures A20 and A21. These can be found in the appendix.

9.1.2 Energy consumption

The total energy consumption of the building having no mechanical ventilation system is another very important issue, although it seems to be secondary when evaluating the chance of taking advantage of natural night ventilation. But when determining the energy use of natural night ventilation it has to be considered that a sufficient amount of outside air has to be provided for hygienical reasons. This outside airflow enters the building through leaks and opened windows or doors and causes a heating load. Having no central mechanical ventilation system means that there is no opportunity in winter to recover any portion of the thermal energy stored in the warm room air leaving the building. This “extra”-heating load has to be taken into account when natural night ventilation is being compared with the reference VAV system.

The building's energy use with natural night ventilation is being calculated considering the 'minimum exchange rate' (compare Figure 9.2) in wintertime. This allows to compare directly the heating energy consumption of the reference VAV system against the building without any mechanical ventilation. In a real building, higher air exchange rates than those assumed are even more likely [39], as people often open windows only a little bit, but keep these opened during the whole working day causing remarkable an air exchange. If the heating system is able to provide enough heating energy to compensate for this, nobody realizes that cold air is entering the room resulting in an increased heating energy consumption.

For the three climates investigated, Figure 9.4 presents the categories of energy use of the building with the reference VAV system on one hand and without a mechanical ventilation and cooling system on the other hand. The energy consumption for the variations without an HVAC system consists just of heating[23] (electricity and fuel), pumps, lighting and equipment.

It is very interesting that the total primary energy use with natural night ventilation for the building exposed to the climate in Berlin is even higher than if the reference VAV system is in operation. If there is no mechanical ventilation system, an significantly increased heating energy consumption occurs, due to the non-existent heat recovery.

Figure 9.4 shows the potential of heat recovery and it has to be taken into consideration that even a favorable user behavior is assumed at this point. As the VAV system applied for Berlin requires just a rather small amount of energy for the chillers and fans, the increased heating energy consumption for the variation with natural night ventilation results in a significantly increased primary energy use of the building (ΔQ_{Berlin} = 11%). The increased heating energy consumption in San Francisco with natural night ventilation is less than the energy saved in the categories "cooling" and "fan", so that the total primary energy consumption decreases by about 7 %. Compared to the reference VAV system, the building expoesed to the climate of Locarno requires about 4 % less primary energy when natural night ventilation is being used (no thermal comfort is provided in this case!).

[23] The rooms are heated just by hydronic baseboards.

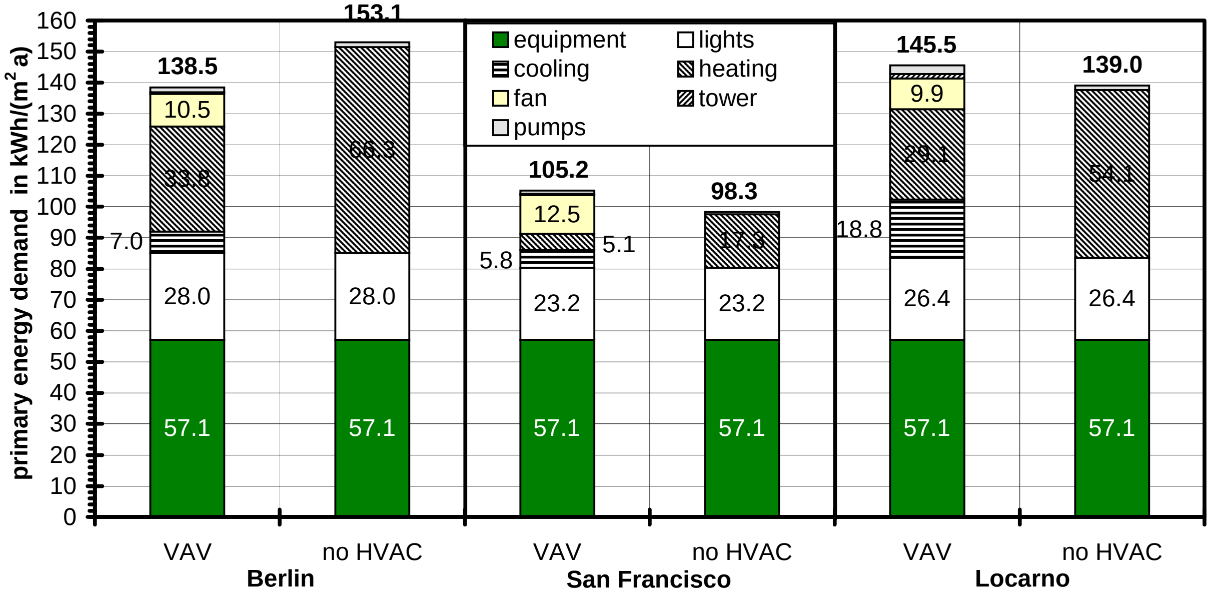

Figure 9.4 : Comparison of the building's primary energy use operating the reference VAV system (VAV) and with natural night ventilation (no HVAC) for Berlin, San Francisco and Locarno

9.1.3 Summary

The results presented show that of the climates investigated just **San Francisco** offers favorable weather conditions for using natural night ventilation energy efficiently. The reference VAV system could be entirely substituted, if the users of the building would be willing to tolerate the temporarily increased humidity levels in the spaces. Mechanical chillers and fans are not required necessarily for cooling the building, which reduces the total electricity consumption by about 19 % and lowers the first costs significantly. The electrical peak power demand drops by 31 % (San Francisco) to 40 % (Berlin) as no compression chiller and no fans need to be operated. However, there is only a little advantage in terms of primary energy saving because the economized electricity is partly compensated by the increased gas consumption for heating. An important positive issue might be the fact that building space can be saved as no duct systems have to be installed between the storeys.

Natural night ventilation can provide comfortable indoor air temperatures in **Berlin**, but the humidity level in the spaces might be too high at almost one third of the cooling season. Although this might probably be tolerable, the primary energy consumption increases significantly compared with the reference VAV system, so natural night ventilation does not seem to be an advantageous strategy for this climate.

There is no realistic way in taking advantage of natural night ventilation considering the climate in **Locarno**. Thermal comfort cannot always be provided sufficiently and only a very little primary energy can be saved when compared with the reference VAV system.

It is really interesting that the primary energy consumption turned out to be the major issue when natural night ventilation is to be judged by its ability to substitute compressive cooling and mechanical ventilation.

Thermal comfort cannot entirely be provided even in moderate climates, as comfortable relative humidity levels throughout the year seem to be out of reach.

9.2 Mechanical night ventilation

Mechanical night ventilation offers the advantage of a controlled supply of conditioned and filtered outside airflow to the spaces. If the night air is vented mechanically, the operation hours of the fans will be longer, unfortunately increasing the building's primary energy use. Additionally, the outside air is slightly heated up by the supply fan (≈ 0.5 - 1.0 K), which reduces the cooling potential of the night air. However, this is the only reliable possibility to supply an adequate amount of outside air and to control the indoor air temperature and the humidity level of the spaces.

Mechanical night ventilation can be accomplished as a additional feature of a conventional HVAC system or as the only cooling method using the potential of the outside air at night presented in the section "Natural Night Ventilation", but providing heat-recovery and reliable supply airflows.

DOE-2E allows to integrate mechanical night ventilation to an already modeled HVAC system with minor effort [52]. The DOE-2E model calculates mechanical night ventilation assuming extra fans for this purpose, but in a real HVAC system the main fans will rather be used. If the intended nighttime airflow rate differs from the supply airflow during the day, a speed-controlled or at least a stepped motor is required.

9.2.1 Mechanical night ventilation without additional cooling

The results of the variations with natural night ventilation showed that the cooling potential of the outside air at night can be sufficient to provide comfortable indoor air temperatures when the building is exposed either to the climates of Berlin or San Francisco. Unfortunately, the outside airflow 'supplied' through opened windows and doors can hardly be controlled or even be guaranteed.

Only mechanically supplying the outside air offers a reliable way of having the intended air exchange rate in each of the spaces independently of the exterior conditions. If temporarily exceeded humidity levels are tolerated, one might consider mechanical night ventilation without an additional cooling coil (and neither a compression chiller nor a cooling tower).

9.2.1.1 Operation strategy

According to the assumed airflow when natural night ventilation was investigated, the airflow rates for the variations with mechanical night ventilation without chiller are set up. During the working hours only the hygienically required outside airflow is being supplied to the spaces (constant airflow). The supply airflow at night was determined considering two issues. First, the pressure drop characteristic of the supply duct with the VAV system is also being used for the mechanical night ventilation with no additional cooling. That means, the same duct size as with the reference VAV system is utilized. Second, the night airflow was adjusted to the required cooling, but the primary energy consumption should not go beyond the primary energy consumption of the reference VAV system. The energy saved for cooling is been used for supplying the outside air at night, therefore, no savings in system's primary energy can be expected.

This kind of set-up does not represent the most energy efficient fan operation. A more energy efficient fan energy use could be achieved, if the ducts were designed according to the higher supply airflow at night. This would provide a lower fan energy use, however, the duct work would require more building's space.

9.2.1.2 Results

Only the final variation of mechanical night ventilation without additional cooling is presented and the indoor air temperatures provided are compared with the results of natural night ventilation and the reference VAV system. The electrical peak power demand can be reduced by 29 % (San Francisco) to 38 % (Berlin) when mechanical night ventilation without chiller is used instead of the conventional VAV system with compression chiller.

Berlin :

The following Figure 9.5 compares the achieved indoor air temperatures of the warmest space of the building when the different night ventilation strategies without chiller are used in a climate like Berlin[24]. The curve for mechanical night ventilation without chiller proves that indoor air temperatures within the comfort range can be provided without any mechanical cooling. The curve for natural night ventilation represents the available cooling potential of the night air. However, this potential cannot be used entirely when the outside air is supplied with fans, as there is always a temperature rise due to the supply air fan. Only at two hours during the very cooling peak day, the indoor air temperature exceeds 26 ^{o}C, but that ought to be acceptable. The relative humidity level inside appeared to be above 60 % at about 200 h and even above 70 % during 18 h.

[24] The results for San Francisco (Figure A27) and Locarno (Figure A26) can be found in the appendix

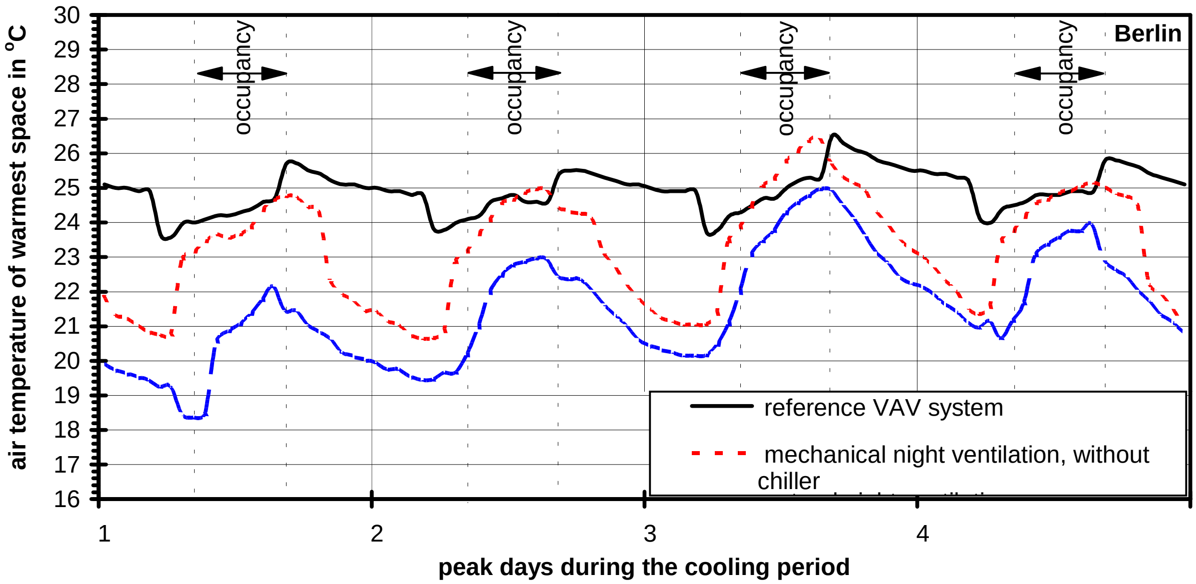

Figure 9.5 : Indoor air temperatures during a cooling peak period for the night ventilation strategies without chiler, compared with the temperatures provided by the reference VAV system (Berlin).

Locarno :

The results for natural night ventilation have already shown that this is not an appropriate cooling strategy with the climate of Locarno. Thus, it is not surprising that mechanical night ventilation without additional cooling is not able to cool the spaces sufficiently (Figure A26, appendix). During the cooling peak period, the outside air at night is not cold enough to offer a sufficient cooling potential. Additional cooling during the day seems to be required in this climate.

San Francisco :

There is one very warm day, which let the night ventilation strategies without additional cooling fail (Figure A27, appendix). Indoor air temperatures of 29 °C are possible, which is definitely uncomfortable. However, as there are only a few hours with temperatures beyond the comfort range, this strategy might be a possible alternative to compressive cooling.

Figure 9.6 compares the primary energy use when mechanical night ventilation without additional cooling is used with the reference VAV system applied to Berlin, Locarno and San Francisco. According to the remarks above, no significant differences in primary energy consumption are possible, but it is a very big advantage, if no compression chiller, no coiling coil and no cooling tower is necessary. This offers remarkable savings with the first and maintenance costs. Some expensive space of the building can be saved as well.

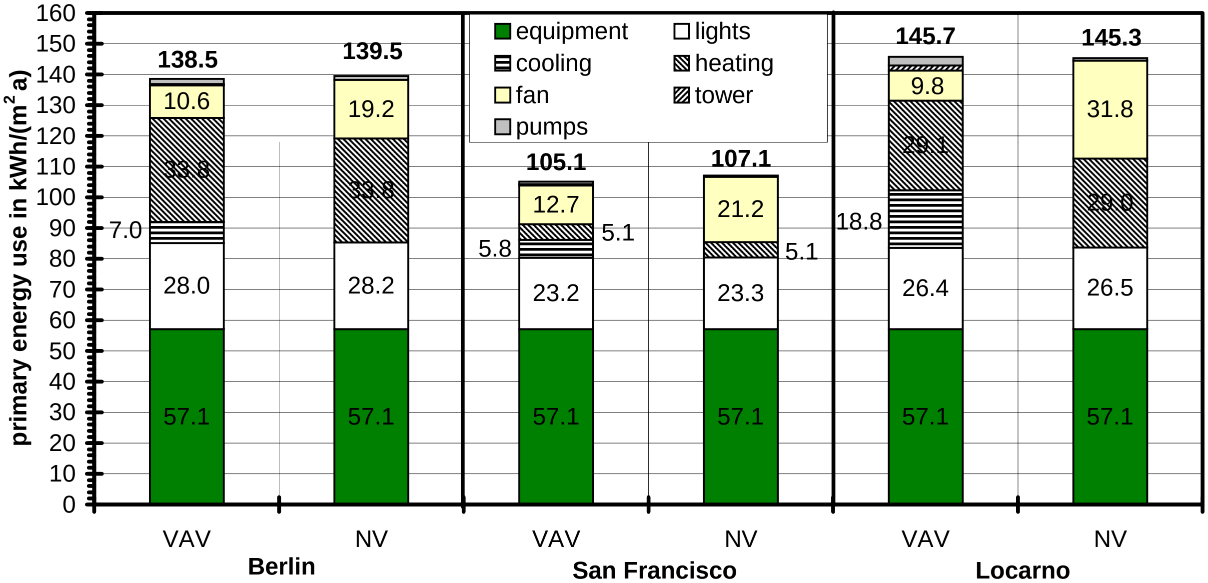

Figure 9.6 : Comparison of the building's primary energy use operating the reference VAV system (VAV) and mechanical night ventilation without additional cooling (NV) for Berlin, San Francisco and Locarno.

9.2.2 Mechanical night ventilation with compression chiller

This night ventilation variation features the conventional VAV system additionally supplying outside at night if there is a potential to pre-cool the building.

9.2.2.1 Cooling strategy

Starting with the configuration of the reference VAV system for each particular climate, night-fan operation has been added for this part of the investigation. The primary energy consumption and the achieved thermal comfort level (indoor air temperature and the relative humidity) are used for evaluating each particular variation. The available cooling capacity, the size of the main fans (supply and return) and the night airflow are adjusted according to the provided comfort level of the preceding variation. Thermal comfort and the energy demand and consumption are compared for several day and night airflow rates

Mechanical night ventilation strategies are examined with main fans and chillers operating during the occupation hours (8^{oo} - 17^{oo} h). Between 22^{oo} - 8^{oo} h, night ventilation fans are only turned on, when the indoor air temperature in at least one of the zones is above 22 ^{o}C and warmer than the outside air temperature[25]. This temperature-controlled operation is provide by DOE-2E

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[25] The temperature rise in the supply air fan is considered.

9.2.2.2 Results

This report presents four slightly different night ventilation variations with additional compression chiller for each of the three climates chosen. These “optimization steps” are supposed to illustrate the influence of the airflow and chiller capacity on thermal comfort and energy consumption. The apparently most energy efficient mechanical night ventilation variations were achieved by varying the supply airflow during night *and* day and by adjusting the available chiller capacity accordingly. This chapter contains detailed charts only for Berlin, summarizing charts and tables for the other climates under study and the discussion of the results.

The detailed charts for Locarno (Figure A22 and Figure A23) and San Francisco (Figure A24 and Figure A25) and a table presenting the characteristic data for the different optimization steps of mechanical night ventilation (Table A19) can be found in the appendix.

Figure 9.7 compares the total primary energy use of the building located in Berlin, with the chiller capacity reduced and additional mechanical night ventilation (NV) used. The remaining cooling capacity is mainly required to dehumidify the supplied outside air, but also to provide the permissible indoor air temperature. Due to the limited cooling capacity, humidity control is not always sufficiently possible, but is being tolerated in these cases in order to leave some energy savings. The following summary discusses the pros and cons of the final mechanical night ventilation versions.

The optimization procedure started with a preliminary fixed airflow for night ventilation of 50 % of the maximum fan capacity (1. and 2. step). At the beginning the maximum fan capacity was reduced according to the changed demand of supply airflow when night ventilation was used additionally. Finally the slightly higher fan capacity[26] of the reference VAV system was used instead, which results in more favorable fan energy use as the pressure drop at lower airflow rates is reduced. A slightly bigger duct than necessary (see footnote 26) does not inevitably lead to increased costs especially in this case, as the reference VAV system is equipped with the same size and represents the basecase.

The impact of the different mechanical night ventilation configurations with chiller operation on the indoor air temperature is shown in Figure 9.8 for Berlin (The curves presented correspond to the columns shown in Figure 9.7). It can be seen that the final version is a compromise of both saving primary energy and providing thermal comfort.

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[26] The set fan capacity in combination with the fixed pressure drop under design conditions determines the characteristic of the duct system (“duct size”). DOE-2 uses this characteristic to calculate the pressure drop for the respective airflow (part load) for each of the operation hours. Increasing the fan capacity and keeping the pressure drop constant causes a more favorable characteristic. For practice, this means that the ducts are sized according to the airflow of the reference VAV system rather than the actual (lower) supply airflow.

At this point it seems to be appropriate to remind the reader that the possible savings of primary energy are rather small, as the reference VAV system is already optimized and represents a very energy efficient system (compare with Chapter 5). If the respectively investigated alternative cooling strategy still offers some energy savings, this definitely justifies the particular strategy, although just relatively low savings can be expected.

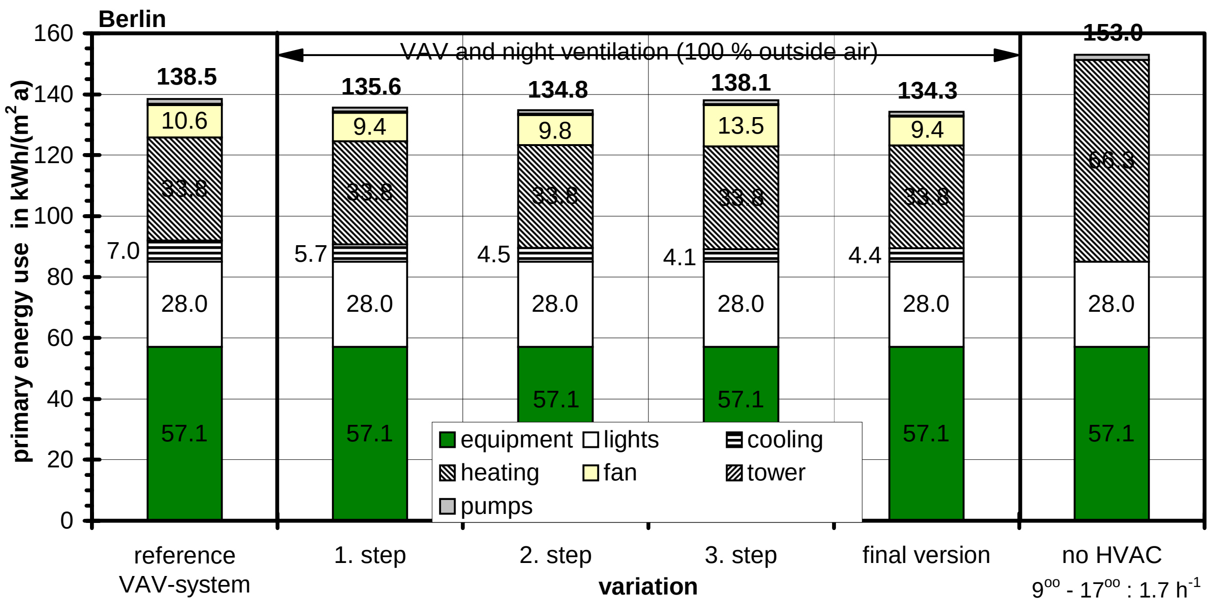

Figure 9.7 : Comparison of the building's primary energy use located in Berlin during the optimization process of mechanical night ventilation with compression chiller.

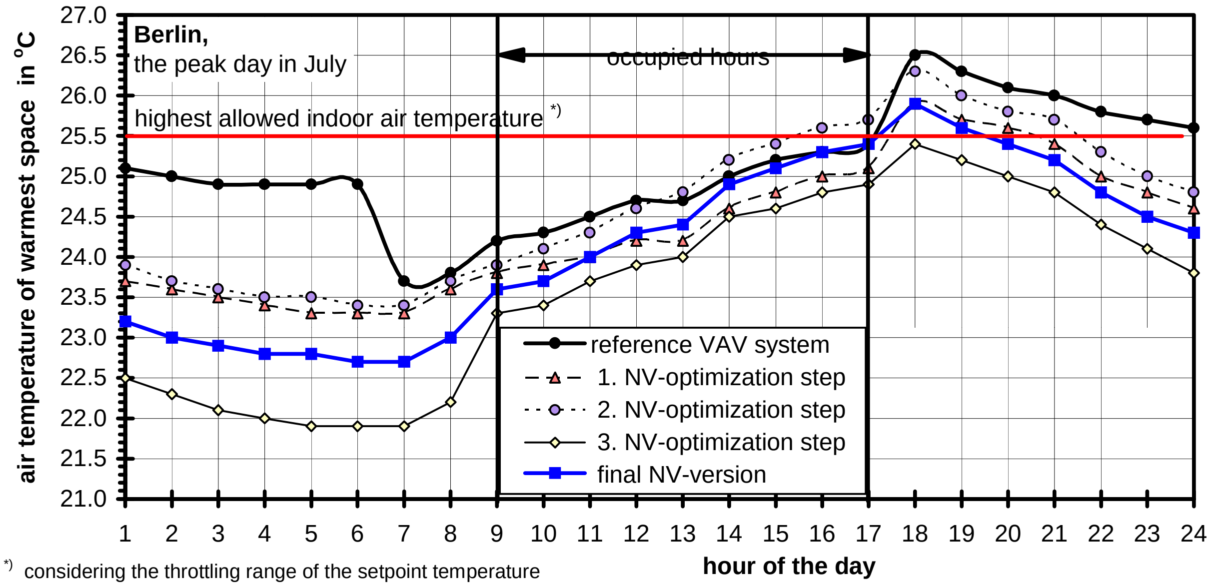

Figure 9.8 : Indoor air temperatures of the warmest space for different mechanical night ventilation configurations with additional compressive cooling in Berlin.

<u>Berlin :</u>

With mechanical night ventilation used in addition to a VAV system with compression chiller, the highest supply airflow during the daytime can be reduced by almost 20 %. During the night, an outside airflow of about 4 $m^3/(h\ m^2)$ is sufficient to obtain a satisfying pre-cooling of the building's structure. This measure decreases the chiller's primary energy use by about 37 % resulting in a reduction of the system's energy use by 8 % when compared with the reference VAV system.

The final set-up of the VAV system with additional, mechanical night ventilation chosen is able to offer very comfortable indoor air temperatures in a climate like Berlin. There are no hours warmer than the permissible temperature. However, the occupants of the building have to tolerate some hours of too high relative humidity levels. The relative humidity in the spaces exceeded 60 % RH at about 50 hours (≅ 3 % of the cooling season). The peak power demand of the system can be reduced as well, the final results for this issue are shown in Table 9.4 (Chapter 9.3).

<u>Locarno :</u>

Providing comfortable indoor air temperatures in the warm and humid climate of Locarno requires the same supply airflow during daytime as the reference VAV system (7400 m^3/h). The airflow at night (4800 $m^3/h \sim 7\ m^3/(h\ m^2)$) is able to pre-cool the building, so that the chiller can be remarkably downsized (20 kW with night ventilation instead of 39 kW for the reference VAV system). The primary energy consumption of the system goes down by about 10 %.

There are just about 30 h of the cooling season (≅ 2 %) with too much moisture inside. However, the rather high moisture content of the outside air in Locarno might involve problems with the indoor air humidity levels. Although, the optimized chiller capacity is almost always able to provide comfortable indoor humidity levels (compare Figure 9.10), night venting the building allows moisture to get stored into the walls, floors, ceiling and other materials, e.g., papers, during the night. Discharging the moisture from the walls might lead to comfort problems during the first hours of chiller operation. As DOE-2E does not consider moisture storage, no reliable information about this problem is available yet.

<u>Red Bluff :</u>

Although, the outside air temperatures at night do not seem to be appropriate for night ventilation, this strategy has been investigated in a preliminary study. Only night ventilation with additional mechanical cooling makes sense in this region, as the outside air temperatures at during the cooling peak period are too warm to provide any benefit.

Supplying outside air at night (8.5 $m^3/(h\ m^2)$) when the temperatures are lower than the indoor temperature reduces the energy use of the compression chiller remarkably. About 10 % of the primary energy consumption of the reference VAV system can be saved. However, as no advantage can be taken of night ventilation at the cooling peak day, the peak power demand of the building remains the same.

<u>San Francisco :</u>

The supply airflow with additional mechanical night ventilation can be reduced slightly. An outside airflow of about 3 $m^3/(h\ m^2)$ is able to pre-cool the building, so that the chiller's capacities as well as the energy use decrease.

At the very peak day of the climate San Francisco, the indoor air temperature exceeds the allowed value. But this happens just for two hours. Considering outdoor temperatures of about 34 ^oC, 27 ^oC in the warmest space appears to be rather acceptable for two hours. All other hours were reported to be within the comfort range. The relative humidity rose above 60 % RH at just 20 h of the working hours (≅ 1 %). Therefore, no basic problems with providing thermal comfort and saving energy occur by using mechanical night ventilation with additional chiller in the temperate climate of San Francisco.

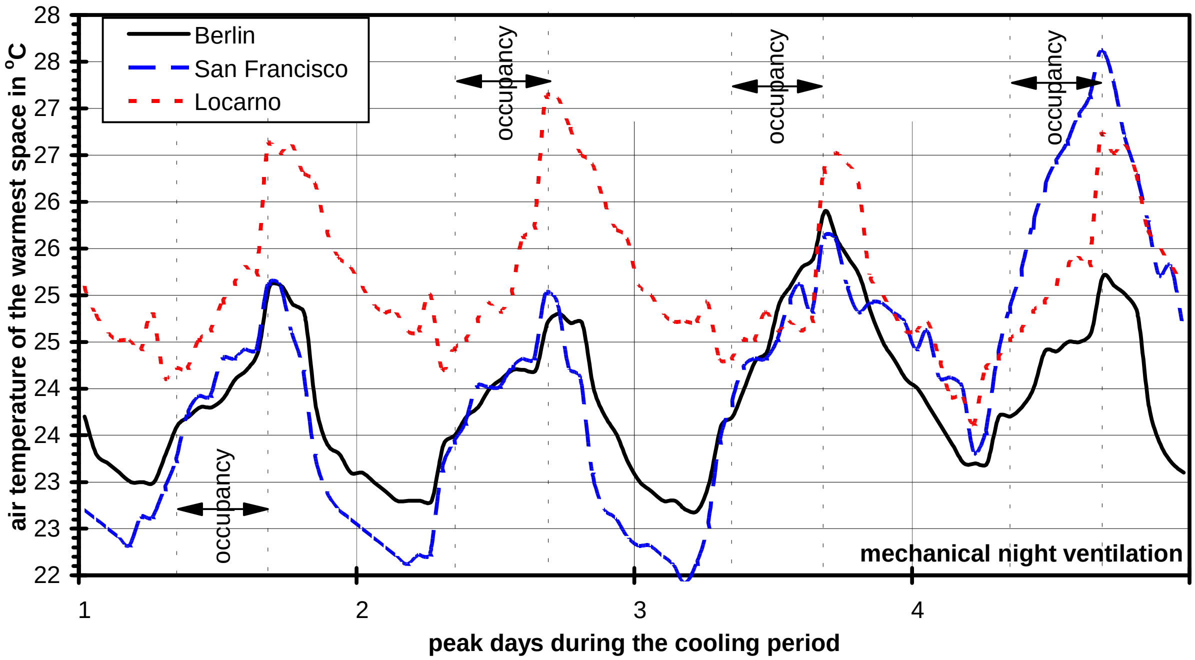

Figure 9.9 : Indoor air temperatures for Berlin, Locarno and San Francisco at four days during the cooling peak period when using mechanical night ventilation and compressive cooling.

Figure 9.10 presents for Berlin, Locarno and San Francisco the indoor air relative humidity levels for four consecutive days during the respective peak period. Only for the climates of Berlin and Locarno some hours during occupancy exceed the permissible humidity level of 60 % RH.

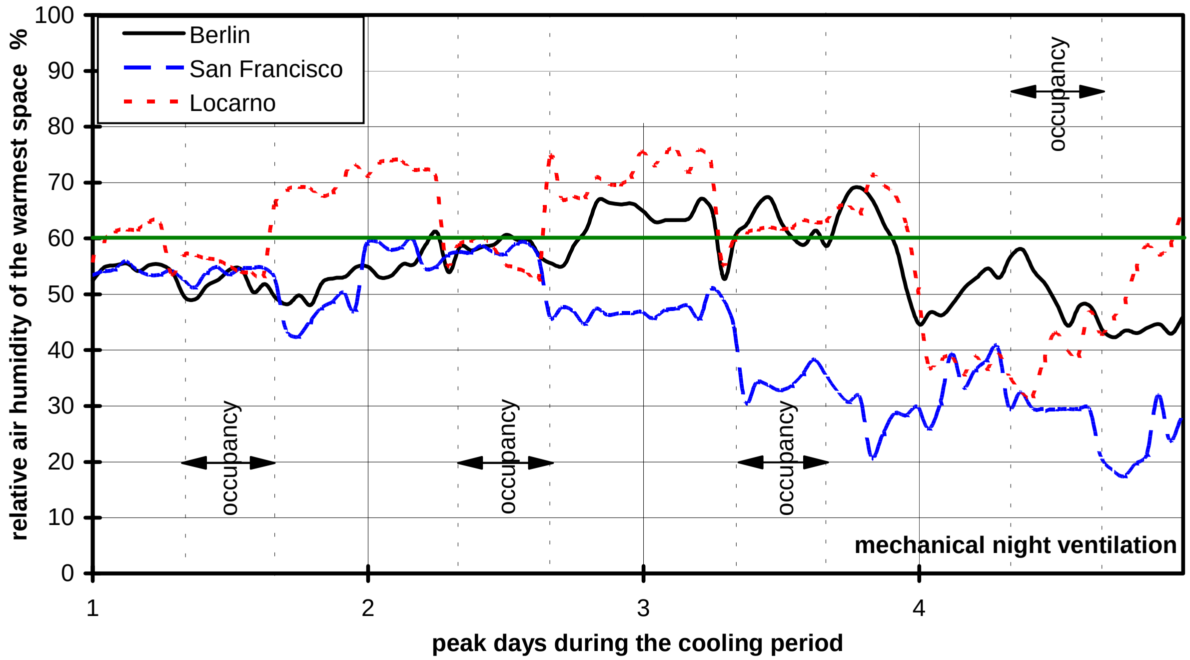

Figure 9.10 : Relative indoor air humidity for Berlin, Locarno and San Francisco at four days during the cooling peak period when using mechanical night ventilation and compressive cooling.

Stetiu et al [46] used the RADCOOL model[27] [45], to investigate among other things the indoor air humidity levels in the same building as being used for this study, but applying very humid climates (New Orleans, Miami). The results of this investigation show that a supplied but unconditioned airflow at night charges a lot of moisture into the building and the humidity levels inside may increase dramatically during the night. In those cases uncomfortable relative humidity occurred during the first hours of occupation. Due to the always existent infiltration with moist outside air, the study concludes that in spite of the necessity of conserving energy, buildings in very humid areas should rather reduce ventilation with *conditioned* air at night instead of turning off the HVAC system entirely.

According to those conclusions [46], it remains uncertain whether the relative humidity in the building, when the climate of Locarno is considered, would be within the comfort range or not. The results presented (Figure 9.10) correspond with non-hygroscopic materials, walls, floors and ceilings, which do not allow any moisture to penetrate their surfaces. An oil-painted surface offers this characteristic for walls, but there are almost always books, papers etc. in a office space, which store moisture and might influence the moisture balance. Thus, using

[27] RADCOOL is being developed to simulate radiation exchange processes in buildings. This model is supposed to make calculations with cooled ceilings possible and it also allows to consider moisture charge and discharge in walls, floors and ceilings.

only the air temperature and humidity ratio during daytime operation provided by DOE-2E, the thermal comfort issue cannot be analyzed without any uncertainties.

Table 9.2 contains the basic data for the most energy efficient mechanical night ventilation strategy (final version).

Table 9.2 : Characteristic design data of the HVAC system using mechanical night ventilation with additional chiller (final version)

variable	unit	Berlin	Locarno	Red Bluff [1)]	San Francisco
$\dot{Q}_{chiller}$	kW	4/6 [2)]	7/13	12/32	4/8
$\dot{V}_{supply,day}$ [3)]	m^3/h	3900	6900	11500	5400
$\dot{V}_{supply,night}$ [4)]	m^3/h	2900	4800	6000	2300

1) : data from a preliminary study
2) : two chillers are used to achieve effective part load operation
3) : daytime fan operation from 9^{oo} to 17^{oo} h
4) : night-fan operation from 22^{oo} - 8^{oo} h (additionally : temperature dependent)

9.2.2.3 Summary

Considering the results presented, mechanical night ventilation with an additional compression chiller appears to be a very competitive alternative to the reference VAV system for the German climate and San Francisco and with some restrictions for Locarno, too. Thermal comfort can be provided and primary energy can be saved, but uncomfortable humidity levels might be possible in the first hours of occupancy in Locarno (see preceding Chapter).

Table 9.3 summarizes the primary energy savings of an HVAC system using optimized mechanical night ventilation with additional cooling over the reference VAV system.

Figure 9.11 compares the primary energy use of the reference VAV system and mechanical night ventilation with additional chiller for the climates investigated by categories. A HVAC system using optimized mechanical night ventilation in addition to the conventional cooling is able to reduce the primary energy consumption of the system by 8 - 10 %, or by 2 - 5 % of the entire building's primary energy consumption (Table 9.3). The required chiller capacity can be reduced by about 25 - 60 %.

Table 9.3 : Primary energy use in % with the final mechanical night ventilation strategy with additional compression chiller when compared with the reference VAV system (= 100 %)

category	Berlin	Locarno	Red Bluff [1)]	San Francisco
chiller	63	70	84	80
heating	100	100	100	100
cooling tower	60	65	83	77
pumps	85	87	94	82
fans	91	102	95	92
HVAC	**92**	**90**	**90**	**90**
building total	97	96	95	98
chiller capacity	**37**	**51**	**100**	**75**

1) : data from a preliminary study

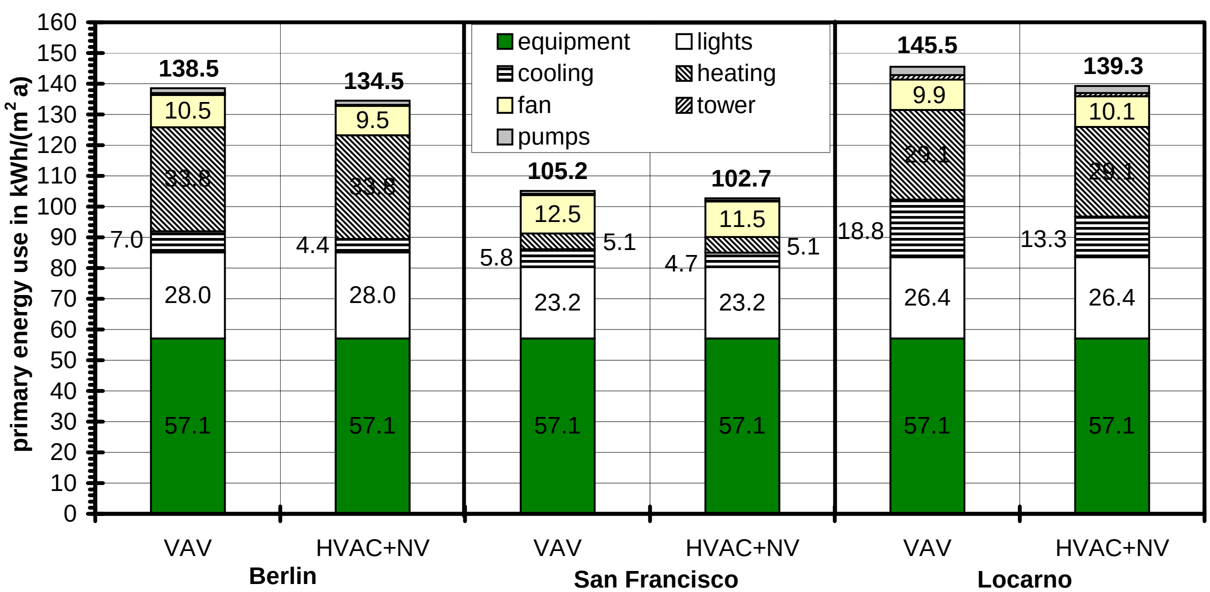

Figure 9.11 : Comparison of the building's primary energy use operating the reference VAV system (VAV) and with optimized mechanical night ventilation with compression chiller (HVAC+NV) for Berlin, San Francisco and Locarno.

9.3 Discussion

Using the colder night air for cooling a building is always a way to reduce the peak cooling load and the required chiller capacity. Additionally, the electrical peak power demand decreases if the outside air temperatures during the cooling peak period are favorable. Depending on the climate, the peak power demand can be reduced by up to 40 % for the variations providing sufficient thermal comfort at most hours of the cooling season.

Table 9.4 : Reduced electrical peak power demand of the building in % for the different night ventilation strategies investigated

		Berlin	Locarno	Red Bluff	San Francisco
no chiller	**natural NV**	-40 [2)]	-52 [1)]	--	-31
	mechanical NV	-38 [2)]	-51 [1)]	--	-29
with chiller	**mechanical NV**	-30	-28	0	-9

1) Room temperatures **and** humidity levels are frequently beyond the thermal comfort range.
2) Indoor air humidity might exceed 60 % RH at about 200 h/a.

The results presented in the preceding paragraphs prove that the night air can be used to sufficiently cool a building with thermal storage capacity only in moderate climates. In some cases, no chiller is required, but then temporarily exceeded humidity levels might affect thermal comfort.

The results for natural night ventilation show the cooling potential of the outside air. But when taking advantage of natural night ventilation, it remains uncertain whether the outside airflow rate assumed can really be reliably ‘supplied’ just by opening the windows and doors. Unfortunately, the increased primary heating energy consumption due to the lack of heat-recovery does not clearly favor natural night ventilation strategy when compared with the reference VAV system. Natural night ventilation seems not to be a competitive alternative to compressive cooling, but in spite of those points mentioned above, a modern office building situated in a moderate climate does theoretically not necessarily need a ventilation system with mechanical cooling to be cooled sufficiently.

Mechanical night ventilation offers controlled air exchange rates, so that the outside airflow becomes independent of the climatic conditions.

If mechanical night ventilation shall be used for thermally condition a building without a chiller, only moderate climates with dry-bulb temperature at night significantly below 20 °C[28] during the cooling peak days should be considered. In

[28] This statement applies just to a type of building and loads used for this investigation and to highest indoor air temperatures of about 26 °C.

such climates, mechanical night ventilation without additional cooling could be an alternative to compressive cooling, which lowers the first and maintenance costs remarkably, but not necessarily the primary energy consumption. When evaluating the results presented, one has to consider that a fan, which increases the air temperature and decreases the cooling potential, supplies the outside air. If the outside air could be 'supplied' to the spaces by using an exhaust fan (vacuum in the rooms) and openings in the exterior walls, even lower indoor air temperatures than achieved with mechanical night ventilation without additional cooling were possible[29]. In the most favorable case, the indoor air temperatures could be as low as with natural night ventilation. However, there is no way of controlling the humidity of the indoor air, which might be an important issue when thermal comfort is top priority.

When mechanical night ventilation is used in addition to compressive cooling, thermal comfort (temperature and humidity) can easily be provided and primary energy savings are possible. The combination of mechanical night ventilation and an entire HVAC system (with compressive cooling) represents a rather conventional way of saving primary energy by taking advantage of 'alternative' cooling sources. This strategy can be introduced to existing systems if speed-variable fans (speed-controlled or stepped) were available and the control system could be adjusted to the changed needs (temperature setpoints and schedules). However, a chiller and a cooling tower, which are usually significantly smaller when compared with a conventional HVAC system without using night ventilation, and cooling coils are still required.

Considering the primary energy consumption, a combination of a 'conventional' HVAC system with mechanical night ventilation seems to be the most favorable and reliable way of taking advantage of the colder night air. This combination also provides good thermal conditions, as the chiller and cooling coil can be used to dehumidify the supply air. This set-up offers savings potentials even in climates with fairly high outside air temperatures at night.

[29] DOE-2E does not provide the opportunity to use only an exhaust fan for night ventilation, thus, this configuration could not be examined.

10. Evaporative Cooling

Evaporative cooling is based on the dry-bulb temperature depression within an air volume when water is evaporated in the air stream. This process is almost adiabatic[30], the wet-bulb temperature of the humidified air stream remains the same and the humidity ratio increases. The air-handling process can be utilized to reduce the supply air temperature either directly or indirectly or both directly and indirectly.

Direct evaporative coolers evaporate water directly into the supply air reducing the dry-bulb temperature while the wet-bulb temperature and the enthalpy are not changed (compare process from point 2 to point 3 in Figure 10.2). Direct evaporative cooling can be perfectly used as a single-stage cooling method for residential buildings in dry (!) climates [25]. The effectiveness of a direct evaporative cooler, η_{dir}, is determined by the difference of the dry-bulb and wet-bulb temperature of the entering air and the performance of the evaporative cooler. The saturation or humidification effectiveness is defined as follows :

$$\eta_{dir} = \frac{\left(t_{dry,enter} - t_{dry,exit}\right)}{\left(t_{dry,enter} - t_{wet,enter}\right)} \tag{2}$$

with : $t_{dry,enter}$: dry-bulb temperature of the entering process air
$t_{dry,exit}$: dry-bulb temperature of the exiting process air
$t_{wet,enter}$: wet-bulb temperature of the entering process air

Spray-type or wetted-media air coolers are used with the saturation effectiveness commonly ranging between 70 and 90 % [1,2]. The effectiveness of modern humidifiers, respectively evaporative coolers, can be controlled continuously down to approximately 20 % by adjusting the pressure of the water evaporated [42]. For this purpose speed-controlled pumps can be used.

An indirect evaporative cooler is a combination of a direct evaporative cooler and an air-to-air heat exchanger. First, the dry-bulb temperature of a secondary air stream is reduced by direct evaporation (compare previous paragraph; see process from point 1 to 6, Figure 10.2). Second, the humidified secondary air extracts sensible heat from the primary air in the heat exchanger (point 6 to 7, respectively point 1 to 2, Figure 10.2).

Indirect evaporative cooling reduces both the dry-bulb and the wet-bulb temperature of the primary air while the humidity ratio remains constant (see $\Delta t_{wet\text{-}bulb}$ and $\Delta t_{dry\text{-}bulb}$; Figure 10.2). The effectiveness of an indirect evaporative cooler ranges commonly between 0.6 and 0.8 and depends on both the saturation effectiveness of the direct evaporative cooler in the secondary air stream and the heat transfer

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[30] The humidification process is adiabatic if the supplied water is neither cooler nor warmer than the air.

effectiveness of the air-to-air heat exchanger. Often rotary heat exchangers with high effectiveness (0.70 - 0.85) and relatively low pressure drop are being used.

Depending on the climatic conditions and the application, combining indirect and direct evaporative coolers might be appropriate to increase the cooling capacity [3]. Outside air, return air or a mixture of these can be used as secondary air, but in order to enable heat recovery with the air-to-air heat exchanger in wintertime, return air should rather be used.

Generally, evaporative cooling can be utilized instead of conventional cooling (cooling coil supplied with cold water from a compressive chiller) if the humidity ratio of the secondary air is lower than or equal to the required humidity ratio of the supply air to provide comfortable relative humidity values in the spaces. As direct evaporative coolers increase the humidity, outside air conditions with a fair amount of moisture are disadvantageous. When the moisture content of the ambient air requires dehumidification an additional cooling coil (supplied with chilled water from a compressive chiller) might be used. Such a three-stage indirect/direct evaporative cooling system (compare Figure 10.1) uses evaporative cooling for pre-cooling the supply air. However, for climates requiring dehumidification frequently, only indirect evaporative pre-cooling is more suitable [33]. Normally, the primary energy use of a building can be reduced with evaporative pre-cooling when compared with a conventional HVAC system.

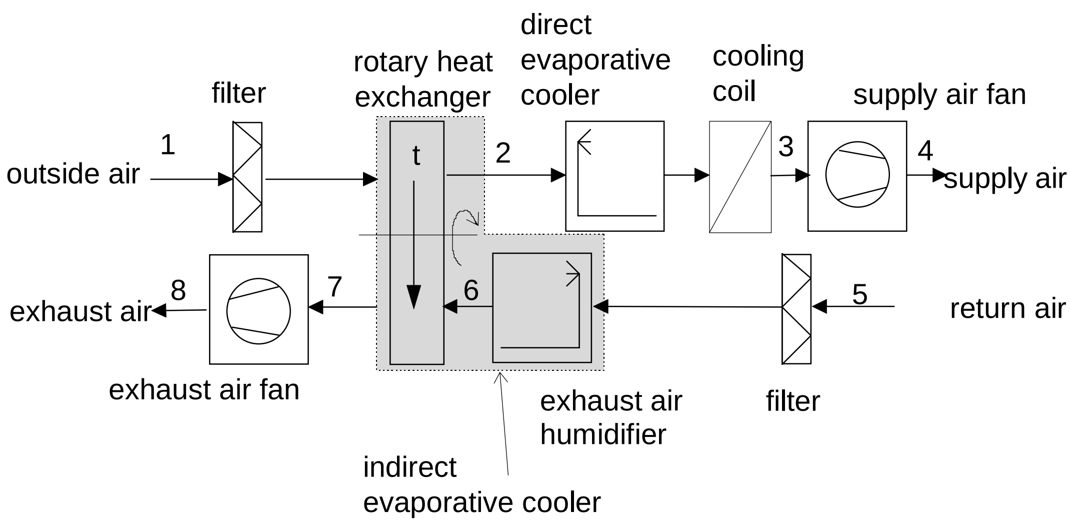

Figure 10.1 : Components of a three stage indirect/direct evaporative cooling unit (heating coils for wintertime are not presented)

Considering the processes of indirect and direct evaporative cooling in the psychrometric chart, advantageous climatic conditions for evaporative cooling can be expected for the Californian climates and to a smaller extent also for the German climates (compare **Error! Reference source not found.**). As the weather data of the German Test-Reference-Years do not differ significantly (compare Chapter 4) for evaporative cooling, only the climate of the TRY 3 is investigated.

Since the temperatures in TRY 3 are slightly higher than those of TRY 1, the more unfavorable boundary conditions are used.

Due to the fairly high humidity ratio of the ambient air in Locarno evaporative cooling does not seem to be adequate for reducing neither the energy consumption nor the cooling capacity of a remaining compression chiller. Nevertheless, a certain peak power reduction can be expected. Direct evaporative cooling certainly does not provide any benefits for this climate. But, since the return air is used as secondary air, the indoor air conditions are influencing the indirect evaporative pre-cooling capability rather than the climatic conditions. Thus, indirect evaporative cooling is investigated using the weather data of Locarno as well.

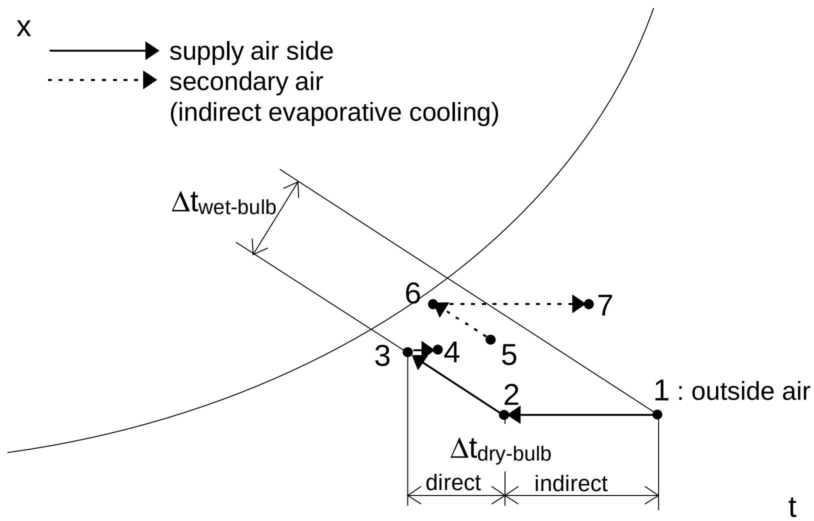

Figure 10.2 : Psychrometric representation of combined indirect and direct evaporative cooling (no cooling coil operated, return air is used as secondary air)

10.1 Set-up for simulating evaporative cooling

DOE-2E provides algorithms to investigate indirect or indirect/direct evaporative cooling systems covering stand-alone and add-on units. The evaporative cooling system investigated is based on ventilation with variable air volume flow. The initial system is using both evaporative cooling and conventional cooling. This set-up makes it possible to control the humidity ratio in the spaces. DOE-2E does not offer a sophisticated humidity control for the air leaving the direct evaporative cooler, but the setpoint temperature of the supply air (here set to 18 ^{o}C) is provided if possible. As a result, some hours with a exceeded humidity ratio (after the direct evaporative cooler) might appear. A cooling coil located downstream of the evaporative cooler is able to dehumidify the supply air if required, controlling the indoor air humidity. Unfortunately this requires reheating to provide the intended supply air conditions. Assuming a continuous humidity control of the saturation effectiveness, this additional heating energy use is not taken into account. However, the dehumidification required to remove the excess water evaporated, increases the cooling energy use slightly. Direct evaporative cooling with a reliable humidity control should obtain even better energy consumption than presented in the following Chapter.

For all variations simulated, the saturation effectiveness of the return air humidifier is set to 90 %[31]. The rotary heat exchanger investigated provides a heat transfer effectiveness of about 75 % for cooling conditions. The pressure drop of the varied HVAC system increases slightly (~11 %) when compared with the reference VAV system. Table A17 (appendix) presents the detailed pressure drops for design conditions used for investigating evaporative cooling.

10.2 Results

Starting with a three stage evaporative cooling system the final system's set-up for each of the investigated climates was found by evaluating both the primary energy consumption and the obtained thermal conditions for the varied operation strategies, the cooling capacity and the supply airflow. Problems with thermal comfort, i.e., with the relative humidity, appear during some hours when direct evaporative cooling is used in climates frequently having outside air humidity ratios above the permissible value for the supply air. With regular latent loads, the humidity ratio of the supply air should not exceed about 10 g/kg. It turned out to be more advantageous to use only an indirect evaporative cooler to pre-cool the ambient air in Locarno as dehumidification is required at many hours during the cooling season. Although, the climatic conditions in Berlin also require dehumidification, direct evaporative cooling increases the pre-cool potential of evaporative cooling. The installed cooling coil, however, provides the additional sensible and latent cooling capacity if required.

The following final set-ups are providing thermal comfort (room temperature and relative humidity) and appear to be the most energy efficient evaporative cooling strategy for the particular climates :

- **Berlin** : three stage evaporative cooling system
- **Red Bluff** : indirect/direct evaporative pre-cooling with conventional cooling coil additionally
- **Locarno** : indirect evaporative pre-cooling with additional conventional cooling coil
- **San Francisco** : indirect/direct evaporative cooling system **without** conventional cooling coil

[31] DOE-2E is adjusting the saturation and heat exchanger effectiveness according to the actual airflow.

Figure 10.3 compares the primary energy use of the evaporative cooling systems with the reference VAV systems for the four climates investigated. Primary energy can be saved in all cases investigated.

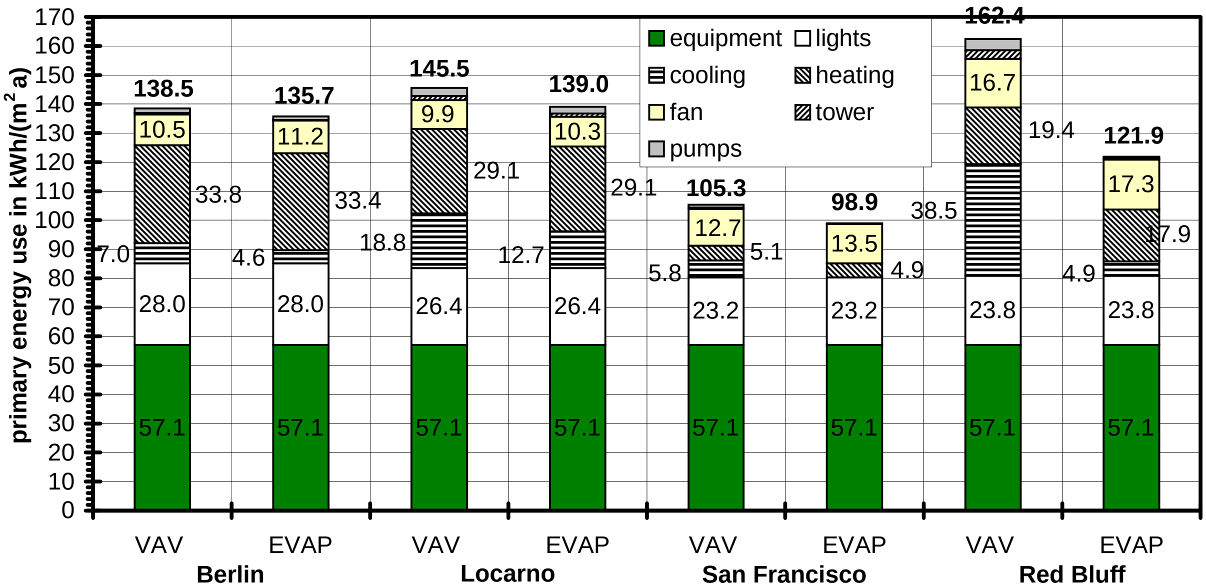

Figure 10.3 : Comparison of building's primary energy use operating either the reference VAV system (VAV) or the optimized evaporative cooling system (EVAP).

Figure 10.5 shows the changes of the primary energy use by category in percent. It also presents the peak cooling load reduced due to the evaporative cooling determining the required chiller capacity. Table 10.3 shows for both the evaporative cooling system and the reference VAV system the hours of operation of the respective cooling devices.

Berlin :

Berlin offers primary energy savings of about 35 % for cooling purposes or about 5 % of HVAC system energy when direct and indirect evaporative pre-cooling is utilized. The overall savings are relatively small as the compression chiller needs to be operated for dehumidifying the outside air and providing the required supply air temperature at about 65 % of the hours when cooling is needed. Some of this dehumidification energy can be saved if a reliable humidification control for the direct evaporative cooler could be applied. The evaporative pre-cooling, however, supports the conventional cooling system, so that the cooling coil, the chiller and the cooling tower sizes can be reduced by almost 40 %. The electrical peak power demand of the building can be reduced by approximately 15 %.

Locarno :

The cooling energy consumption in Locarno decreases by about 30 % when an indirect evaporative pre-cooler is used. This results in savings of about 10 % in HVAC energy. Unfortunately, the peak cooling load remains almost the same as it occurs during a very humid (17.7 g/kg) and warm (34 ^{o}C) hour when the indirect evaporative pre-cooler provides only a small cooling potential. The peak power demand does not change either.

Red Bluff :

The most impressive primary energy savings occur in the very dry and hot climate of Red Bluff (Figure 10.5). Pre-cooling the supply airflow by means of indirect and direct evaporative cooling reduces the chiller energy use (third stage) by more than 85 % and the system's primary energy consumption decreases by 50 %. The compression chiller needs to run only at about 20 % of the hours with a cooling load . The required compressive chiller can also be significantly downsized as the peak cooling load for the cooling coil is reduced by about 50 %. The electrical peak power demand of the HVAC system can be reduced to about 60 % of that of the reference VAV system (Table 10.1).

Surprisingly, the evaporative cooling set-up requires less heating energy than with conventional cooling in that dry and hot climate (see Figure 10.3). This happens only due to reheating operated during the cooling season !

To explain this issue, the following Figure 10.4 presents the air-handling processes for these two different cooling methods for one particular hour in July in Red Bluff.

The conventional system (———) cools the very hot and dry supply air (1) down to the setpoint temperature (2). Then, the supply air is humidified to achieve a relative humidity inside the comfort range (3). This humidification decreases the dry-bulb temperature of the supply air below the setpoint requiring reheating (4). Better energy efficiency could be achieved with a changed control strategy. Anyway, this is the common way of handling air during the cooling period but, it has to be stated that it is not quite common to have humidification needs in summer. Usually, the supply air has to be **de**humidified which, however, might also require reheating. Supply air temperatures below the setpoint can occur while dehumidification as the humidity ratio of the supply air has priority over the temperature achieved. To dehumidify the supply air sufficiently, fairly low chilled water temperatures are required resulting in low supply air temperatures. Depending on the design of the system, reheating might be necessary to provide the right supply air temperature.

The evaporative cooling system (----------) does not require reheating energy as the supply air setpoint is approached by utilizing the direct evaporative cooler (2a to 3a). The control of the direct evaporative cooler humidifies the supply air as long as the intended supply air temperature is provided.

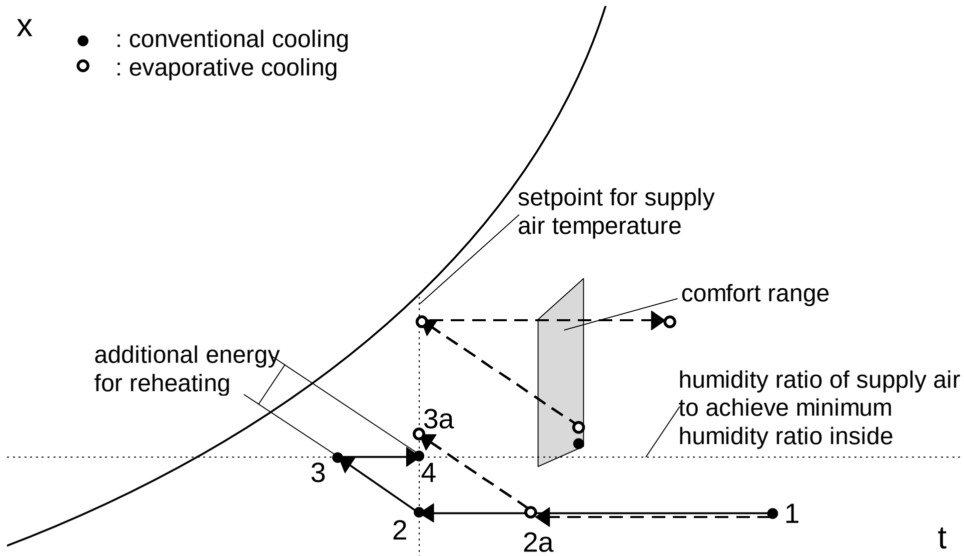

Figure 10.4 : Psychrometric representation of the air-handling processes with conventional and evaporative cooling for the dry and hot climate of Red Bluff

Although, a combination of evaporative and conventional cooling appears to be the most energy efficient set-up with evaporative coolers for Red Bluff, evaporative cooling without any additional conventional cooling is also able to provide sufficient thermal comfort. Due to the extremely warm outside air in summer, the supply air temperature possible, unfortunately, is often higher than the intended setpoint. Therefore, the supply airflow needs to be increased significantly to provide enough cooling capacity. As a result, the increased fan energy consumption and fan power demand[32] do not favor this possibility over the one with additional compressive cooling. However, taking into account that neither a chiller, nor a cooling tower or cooling coil is required in this case, the less compelling savings possible might be overruled by the more advantageous first costs and maintenance. The two variations of evaporative cooling investigated are compared in Table 10.1.

For the climate of Red Bluff, Figure 10.5, Table 10.2 and Table 10.3 are presenting only the results for the variation with evaporative coolers and compression chiller.

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[32] The fan energy use and power demand for the stand-alone system was calculated assuming the same size of ducts as for the reference VAV system and the system with evaporative coolers and compression chiller. If the ducts were sized according to the increased airflow, the evaporative cooling system would have operated more energy efficient.

However, as the airflow had to be increased by about 50 % the duct size would have to be increased accordingly, requiring significantly more space for the duct work and the HVAC unit.

Table 10.1 : Primary energy use and electrical peak power demand of evaporative cooling strategies in % for the climate of Red Bluff, compared with conventional compressive cooling (reference VAV system = 100 % for each of the categories)

	annual primary energy consumption				peak power demand	
HVAC set-up	cooling + tower	fans	HVAC	building	HVAC	building
indir. + dir. evap. cooling	0	258	**75**	87	**97**	96
indir., dir. evaporative + compressive cooling	22	103	**50**	75	**60**	70

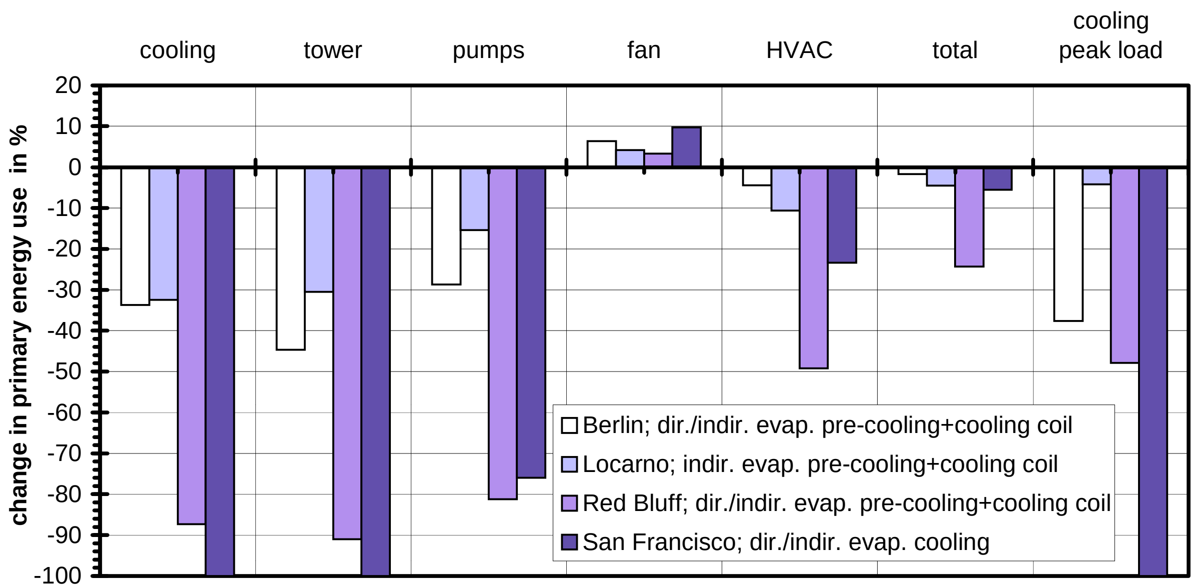

Figure 10.5 : Changes in primary energy use when the final evaporative cooling system for each of the climates investigated is compared with the corresponding reference VAV system.

<u>San Francisco :</u>

The indirect/direct evaporative cooling system in San Francisco does not need to have an additional conventional cooling coil as no dehumidification is required and the outside air temperatures are moderate. Due to the increased pressure drop of the unit, the fan energy use is about 10 % higher than with the conventional system. Nevertheless, the system's primary energy consumption can be reduced by about 24 %. However, due to the fact that the reference VAV system in this climate consumes only about 24 % of the building's primary energy use (compare Chapter 8.4.2), the most impressive savings may appear in first costs.

The electrical peak power demand of the HVAC system can be reduced by up to 40 % when comparing with the reference VAV system with compression chiller.

To provide indoor air conditions within the comfort range, the effectiveness of the direct evaporative cooler needs to be controlled continuously. However, during most of the cooling season a saturation effectiveness of about 40 - 50 % seems to be sufficient in this particular climate and was used for the results presented. As the direct evaporative cooler is also used for humidification in wintertime when a high effectiveness (about 90 %) is desired, either a controllable spray-type humidifier [42] or two different operation modes, e.g., two separate banks of spray nozzles, are required.

Table 10.2 shows the peak power demand with the respective evaporative cooling system compared to the reference VAV system.

Table 10.2 : Electrical peak power demand in % of the evaporation cooling set-up, compared to the reference VAV system (= 100 %).

	location			
category	**Berlin**	**Locarno**	**Red Bluff**	**San Francisco**
HVAC in % [1)]	72	100	60	21
entire building in % [2)]	84	100	70	75

1) The category HVAC contains the energy use of cooling, heating, pumps, tower and fans.
2) The building's power demand includes all categories, i.e., lights and equipment, too.

Table 10.3 : Operation hours of the cooling devices for both the evaporative cooling systems and the reference VAV systems for the climates investigated.

	evaporative cooling					**conventional reference VAV system**	
	evap. coolers operated		**chiller operation**				
location	h	% [a)]	h	% [a)]	% [b)]	h	% [a)]
Berlin	645	38	415	24	64	630	37
Locarno	1055	62	895	53	85	1080	64
Red Bluff	1435	84	235	14	16	1465	86
Red Bluff evaporative cooling only	1498	88	no chiller operated			1465	86
San Francisco	880	52	0	0	0	910	54

a) Refers to the operation hours of the fans during the cooling season (April - October ≈ 1700 h).
Weekends and holidays are considered as well. There are 1694 h during this time.
b) Refers to the hours of operating the evaporative cooling system.
*Example Red Bluff : The evaporative coolers are operated during 1435 h and the compression chiller is running at 233 h additionally. (233 h/1435 h)*100 ≈ 16 %.*

10.3 Summary

Especially in the dry Californian locations investigated evaporative cooling offers remarkable saving potentials. A stand-alone evaporative cooling system is able to cope with the cooling loads occurring in the building investigated.

In San Francisco, significant savings in first costs and to a lesser extent in primary energy consumption can be achieved with stand-alone evaporative cooling. This is the most desirable application for evaporative cooling as no compression chiller is required. However, San Francisco offers only small absolute primary energy savings as the reference VAV system demands just about one quarter of the entire building's energy. Thus, the savings in primary energy are not compelling but the opportunity to cool the building sufficiently without a compressor-driven chiller is very attractive.

When the building is exposed to the hot and dry climate of Red Bluff the most energy efficient set-up is a combination of evaporative pre-cooling and conventional, compressive cooling. The primary energy consumption, the peak cooling load remaining for the compression chiller and the building's electricity demand can be reduced by half when the supply air is pre-cooled with indirect and direct evaporative coolers. Compared with conventional HVAC systems having a heat-recovery unit and using 100 % outside air, savings in first costs and in maintenance are very likely, since there is only one additional humidifier needed in the return air but the chiller and cooling tower capacities required are much smaller (compare Figure 10.5). For the climatic conditions of Red Bluff the stand-alone evaporative system offers only lower savings in primary energy use and no peak power reduction. Anyway, the building can be thermally conditioned without a compression chiller and cooling tower.

For the European climates investigated evaporative pre-cooling offers savings in primary energy for the HVAC system of about 5 % (TRY 3) to 10 % (Locarno). Compression cooling is required for dehumidification, but the cooling capacity can be reduced significantly. For the moderate German climates a peak cooling load reduction of almost 30 % can also be achieved. The peak power demand of the HVAC system can also be lowered by about 30 %. In Locarno, the chiller can not be downsized, so that the peak power demand remains the same.

The water consumption of the evaporative coolers is usually smaller than that of a cooling tower of a convetional system, since the physical principle is the same[33] but the cooling tower has to reject both the heat (cooling load) from the conditioned spaces and additionally the energy (electrical input) of the compressor which makes up about 25 - 30 % of the heat removed in the cooling coil.

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[33] The boundary conditions are not exactly the same for both cases as cooling towers are always using outside air to cool the cold water while the indirect evaporative cooler is evaporating water in the return air stream, having different properties when compared with the outside air.

11. Desiccant Cooling

The investigation of the previous cooling strategies has shown that the latent load restricts alternative cooling methods rather than the sensible load. As the moisture content of the supply air affects thermal comfort in the spaces dehumidification is required during the cooling season in regions like the European climates investigated.

Conventional HVAC systems remove moisture from the supply air, respectively the latent load from a space, by cooling the supply air below the dewpoint temperature. This results in dehumidification of the supply air due to condensation at the cooling coil pipes. This kind of dehumidification usually requires coolant temperatures of about 6 - 12 °C, making mechanical cooling inevitable. Chillers can provide these temperatures without problems, although the energy use increases with decreasing chilled water temperatures. Alternative cooling strategies usually cannot provide such temperatures, thus, dehumidification by means of cooling below the dewpoint is not possible[34].

However, instead of using mechanical cooling which normally implies a high power demand by the compression chiller, the supply air can also be dehumidified with sorptive materials, called desiccants. Taking advantage of a continuous and regenerative process, makes it possible to control the indoor air humidity without using CFC, HCFC or HFC refrigerants which are harmful to the ozone layer. The materials used for desiccant cooling are neither hazardous to the environment nor toxic to people. The electric power demand of such a system drops remarkably when compared with a conventional HVAC system utilizing compressor-driven cooling.

The first costs of a desiccant cooling system might be as high as for a conventional HVAC system with compression chiller and cooling tower. Depending on the size of the unit, even more favorable first costs are possible [10,42]. All the components utilized are highly developed and have been used for many years for air-conditioning or other air-handling processes.

11.1 Principle of operation

In fact, desiccant cooling systems are a combination of an adsorptive or absorptive dehumidifier and indirect and direct evaporative coolers (compare Chapter 10). For air-condition purposes, rotary desiccant dehumidification units (‘desiccant wheel’) which inner surfaces are covered with a solid desiccant, e.g., silica gel, are commonly used. Figure 11.1 depicts such a desiccant cooling system by its components schematically and Figure 11.2 illustrates the corresponding processes in the psychrometric chart.

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[34] An exception is the absorption chiller which is able to provide the same temperatures as compression chillers. Using absorption chillers will be investigated in Chapter 12)

Instead of using mechanical energy to cool and dehumidify the supply air with a compression chiller or cooling coil, respectively, desiccants (liquid or solid) are used to attract and retain the moisture in the first stage (latent load) of the desiccant cooling system. The dried supply air leaves the dehumidification wheel warmer as sorption heat is released (Figure 11.2 : process from 1 to 2). An indirect evaporative cooler represents the second stage and either return air or outside air can be used as secondary air to remove sensible heat from the supply air in the heat exchanger (process from 2 to 3). Finally, a direct evaporative cooler decreases the dry-bulb temperature of the supply air (third stage; process from 3 to 4).

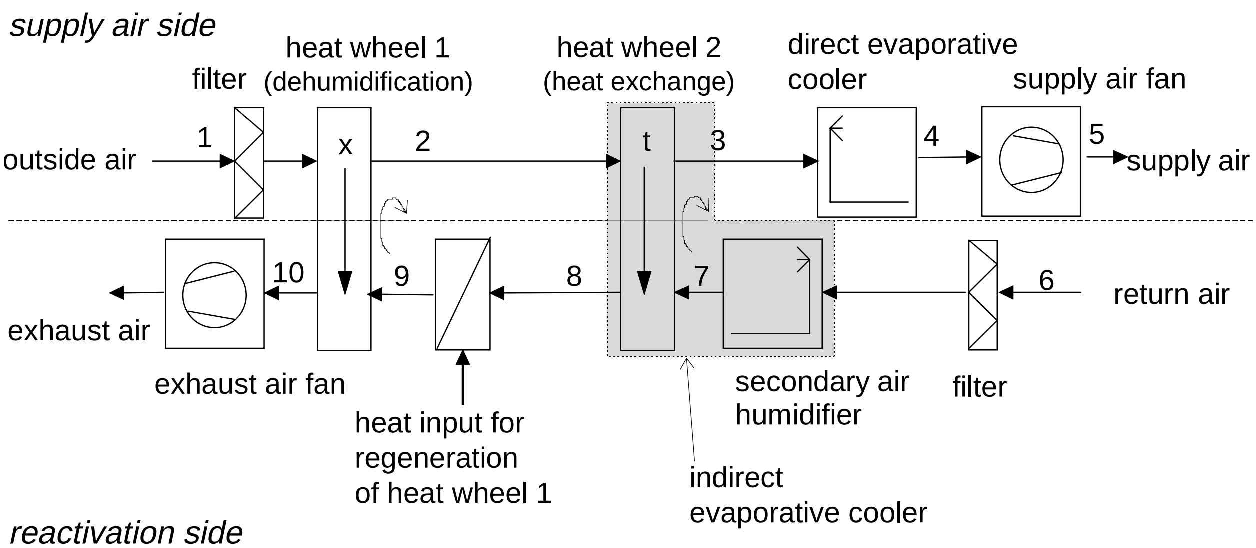

Figure 11.1 : Schematic of an air handling unit with rotary solid desiccant dehumidifier, indirect and direct evaporative cooler (using return air in the indirect evaporative cooler and for regeneration).

Blowing heated return air or outside air through the desiccant wheel (process from 9 to 10 regenerates the desiccant loaded with moisture). To achieve an appropriate dehumidification, the relative humidity of the entering regeneration air has to be at least as low as that of the supply air exiting the desiccant wheel [9,12]. Depending on the required temperature level for the regeneration, gas- or oil-fired heaters, waste heat or heat pumps [23] can be used to provide the heat to dry and regenerate the desiccant continuously (process from 8 to 9).

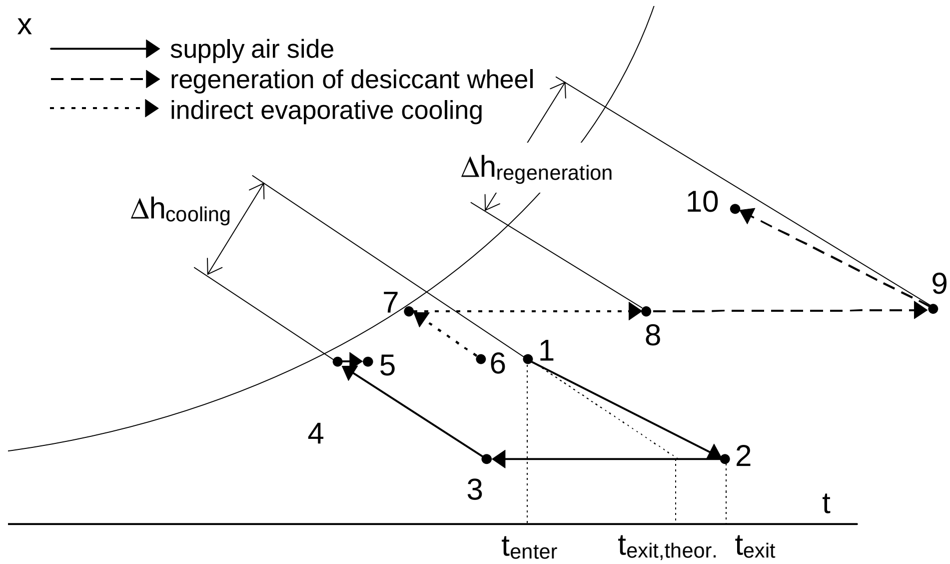

Figure 11.2 : Psychrometric representation of a cooling process with an air handling unit with rotary solid desiccant dehumidifier and direct and indirect evaporative cooling (using return air in the indirect evaporative cooler and for regeneration)

The most common desiccant cooling system for air-conditioning buildings is a combination of a rotary air-to-air heat exchanger covered with a solid desiccant, e.g., silica gel, a heater (gas-fired or water-heated) for regenerating the desiccant and both an indirect evaporative cooler and a direct evaporative cooler [12]. The desiccant wheel can be divided in two or three sections. The moist process air (supply air) is drawn through the first section. The heated secondary air to remove the moisture absorbed in the first section flows through the second section regenerating the wheel. A purge airflow (outside air) might be used to penetrate a small third section between the first and the second section in order to cool the wheel and to reduce unwanted heat transfer from the regeneration section to the process air section.

The process presented above features the temperature and humidity changes when conditioned air from the spaces (return air) is used on the reactivation side and 100 % outside air is supplied to the spaces. However, return air might be mixed with the outside air or only outside air can be used both to regenerate the desiccant wheel and to cool the supply air via the indirect evaporative cooler. Depending on the climatic parameters and the particular cooling loads, the one or the other air stream might be the most favorable. The air stream with the lowest wet-bulb temperature incorporates the most advantageous condition when being used as secondary air in the indirect evaporative cooler. For regeneration purposes, usually, the air stream with the highest dry-bulb temperature is the best. However, to take advantage of the existing heat exchangers in wintertime (heat recovery) return air needs to be used as secondary air. This is the common set-up for desiccant cooling systems for air conditioning purposes currently installed in Germany.

The efficiency of a desiccant cooling system (DCS) can be evaluated by the COP (coefficient of performance) which is the ratio of the cooling capacity required to cool the outside air down to the supply air conditions, $\dot{Q}_{cooling}$, and the heat input needed for regeneration, $\dot{Q}_{pr.regen}$ [35]:

$$COP_{DCS} = \frac{\dot{Q}_{cooling}}{\dot{Q}_{pr.regen} + \dot{Q}_{pr.evap}} = \frac{COP_{heater} \cdot \left(\dot{m}_{supply} \cdot \Delta h_{cooling}\right)}{\left(\dot{m}_{regeneration} \cdot \Delta h_{regeneration}\right) + \dot{Q}_{pr.evap}} \qquad (3)$$

with :

$\dot{Q}_{cooling}$	: cooling energy to reduce the outside air temperature to the supply air setpoint temperature
$\dot{Q}_{pr.regen}$	: primary energy consumed for regeneration
$\dot{Q}_{pr.evap}$	: primary energy consumed by the evaporative coolers
COP_{heater}	: effectiveness of the, e.g., gas-fired, heater for regeneration purposes
$\dot{m}_{supply}$	: massflow of the supply air
$\dot{m}_{regeneration}$	: massflow of the regeneration air
$\Delta h_{cooling}$	: enthalpy difference between outside and supply air
$\Delta h_{regeneration}$	: enthalpy rise in the heater for regeneration

Only if the massflow of the supply air, $\dot{m}_{supply}$, and the regeneration air, $\dot{m}_{regeneration}$, are the same, the COP value of the heater (regeneration) equals 1 and the pump energy use of the evaporative coolers can be neglected when compared with the regeneration energy consumed, the differentials of the specific enthalpy shown in the psychrometric chart might be used to determine the COP value of the system at a particular hour (compare Figure 11.2).

$$COP_{DCS} = \frac{\Delta h_{cooling}}{\Delta h_{regeneration}} \qquad (\text{with}: \dot{m}_{supply} = \dot{m}_{regeneration}) \qquad (4)$$

11.2 The desiccant cooling model

The building simulation tool DOE-2E provides a model for simulating desiccant cooling as an addition to a conventional HVAC-system. Several different system

[35] If the desiccant is regenerated by a gas-fired heater $\dot{Q}_{regeneration}$ is represented by the gas consumption.

set-ups are available but unfortunately, the model was created to be used for residential applications (single homes) rather than for commercial buildings. As a consequence, the existing DOE-2E model is not able to calculate reasonable results with the usual airflow rates for large buildings which are usually much higher than those in residential buildings. Thus, the original DOE-2E model could not be used for this investigation.

Stephen Kemp with the Technical University of Nova Scotia in Halifax, Canada, and the Energy Diversification Research Laboratory (CANMET) in Quebec, Canada, have been working on a simulation tool for desiccant cooling and they have created their own desiccant cooling model. This model is designed to be used in conjunction with DOE-2E[36] and is able to handle desiccant and indirect and direct evaporative cooling for commercial buildings. The model is still being developed and has been used for this investigation with approval from CANMET. The model consists of two functions to be implemented into the DOE-2E-input file, thus, no change of the DOE-2 source code is required.

As CANMET and the Technical University of Nova Scotia respectively, had different major points of interest, the model had to be modified for this study. The most significant changes involve the air being used for regenerating the desiccant wheel and in the indirect evaporative cooler on the one hand, and the performance (moisture removal and temperature rise) of the desiccant wheel on the other hand. The final version of the desiccant model used differs significantly from the “Canadian” model.

The most important characteristics of the desiccant cooling model used are:

- 100 % outside air is supplied to the spaces for all variations (no recirculation air). Return air is used both for regeneration and in the indirect evaporative cooler[37] (supply airflow = return airflow). No purge airflow is considered.
- The performance of the desiccant wheel implemented is defined by using data for one particular type of wheel (MUNTERS MCC-1). The performance is based on manufacturer’s data for regeneration temperatures of 75 and 95 ^{o}C. Linear interpolation and extrapolation ($t_{regen,min} \approx 60\ ^{o}$C; $t_{regen,max} \approx 115\ ^{o}$C) is used to determine the performance of the wheel for other temperatures. (compare Chapter 17.6, Appendix).
- A floating regeneration temperature is used. The temperature provided is adapted to the required moisture removal and the outside air conditions. (see also Chapter 17.6.1, Appendix).

[36] The development of the simulation model has been a task of the IEA Annex 28, “Low Energy Cooling”.

[37] The original model was set up to use outside air for both the indirect evaporative cooler and regeneration.

- Both the direct evaporative cooler and the ‘humidifier’ in the indirect evaporative cooler have a fixed saturation effectiveness of 90 %. There is no control of the saturation effectiveness.
- The rotary heat exchanger (indirect evaporative cooler) has a heat exchanger effectiveness of 75 % in wintertime and 85 % in summertime[38].
- A gas-fired heater regenerates the wheel (heater efficiency : 90 %). Using waste heat for regenerating the wheel is being discussed in Chapter 11.3.2 “Regeneration with waste heat”.
- With desiccant cooling, the supply airflow is adjusted to the cooling loads (VAV system)
- The supply air temperature is set to 18 ^{o}C (cooling season), the permissible relative indoor air humidity is 60 % for the cooling season.
- The pressure drop of the entire system increases from 1100 Pa (reference VAV system; Table A16) to 1660 Pa (desiccant cooling system; see Table A18)

11.3 Results

First of all, it turned out that the dehumidification process in the desiccant wheel is far away from being adiabatic. Not only the condensation heat but also the heat of sorption rises the temperature of the air leaving the desiccant. In addition, heat transfer from the warmer regeneration side to the process air takes place. This characteristic is not only valid for the particular sorptive heat exchanger used in the model but, is apparently a typical feature as the characteristic data from different manufacturers of desiccant wheels show all the same behavior (compare Figure 11.2).

As many papers dealing with desiccant cooling mention that the air-handling process in the desiccant wheel is almost adiabatic, it seems to be appropriate to actually quantify the deviation of the dehumidification process in the desiccant wheel modeled from the adiabatic process. The increase of enthalpy depends mainly on the desiccant material (heat of sorption[39]) and the heat transferred from the regeneration air to the process air. Introducing a specific temperature difference, $\Delta t_{desiccant}$, this deviation can be determined (compare Figure 11.2) :

$$\Delta t_{desiccant} = \frac{t_{exit} - t_{enter}}{t_{exit,adiabatic} - t_{enter}} \tag{5}$$

[38] These values are based on manufacturer’s data. The total heat exchanger effectiveness in wintertime increases to about 85 %, as the desiccant wheel contributes to the heat recovery.

[39] For the commonly used desiccant silica-gel adsorbing water vapor, the heat of sorption makes up about 10 % of the condensation heat of water vapor.

with : t_{enter} : dry-bulb temperature entering the desiccant wheel
t_{exit} : dry-bulb temperature leaving the desiccant wheel
$t_{exit,theor.}$: theoretical exiting temperature when the dehumidification process was adiabatic

The simulation runs for different outside air conditions showed temperature rises within the desiccant wheel (t_{exit} - t_{enter}) of about 3.0 - 3.5 K per g/kg moisture removed. An adiabatic dehumidification process, if it was possible, would result in a increase of dry-bulb temperature of about 2.5 K per g water vapor removed. Including the heat of sorption increases this value to about 2.8 K/g. Considering these numbers, the higher results derived by means of simulation lead to values for $\Delta t_{desiccant}$ of approximately 1.2 to 1.4.

Desiccant cooling is only investigated for the European climates Berlin (German TRY 3) and Locarno. As seen in the preceding chapter, the climatic conditions in Red Bluff and San Francisco do not require dehumidification during the summer, hence, desiccant cooling does not seem to be a wise method here (compare Chapter 10)

The energy consumption of the motors to rotate both the adsorption wheel and the rotary heat exchanger, as well as, the energy use of the pumps to spray the water in the evaporative coolers are estimated as this part of the simulation model is not yet completed. Based on the hours of operation and an estimated electrical input, the annual energy use of these components is calculated. Table 11.1 presents the data for these and the results of this estimate.

Table 11.1 : Primary energy consumption of the rotary heat exchangers and evaporative cooler pumps

		Berlin			Locarno		
	type of device	$h_{operation}$	N_{input}	Q_{annual} 1)	$h_{operation}$	N_{input}	Q_{annual} 1)
component		h/a	W	kWh/m^2 a	h/a	W	kWh/m^2 a
desiccant wheel	motor	276	100	0.11	478	100	0.20
indirect evap. cooler	motor	523	100	0.21	998	100	0.41
	pump	523	100	0.21	998	100	0.41
dir. evap. cooler	pump	445	100	0.18	732	100	0.30

1) : This is the estimated primary energy use considering the conversation efficiency of generation of electricity (35 %) and the gross floor area of the building

11.3.1 Regeneration with gas-fired heater

First, desiccant cooling is simulated considering a gas-fired heater for regeneration purposes. Later, the saving potentials with waste and district heat usage are estimated based on the results with the gas-fired heater (Chapter 11.3.2 and 11.3.3, respectively).

The primary energy consumption of the simulated gas-fired desiccant cooling system is compared with the reference VAV system (compare Chapter 8.4.2) in Figure 11.3.

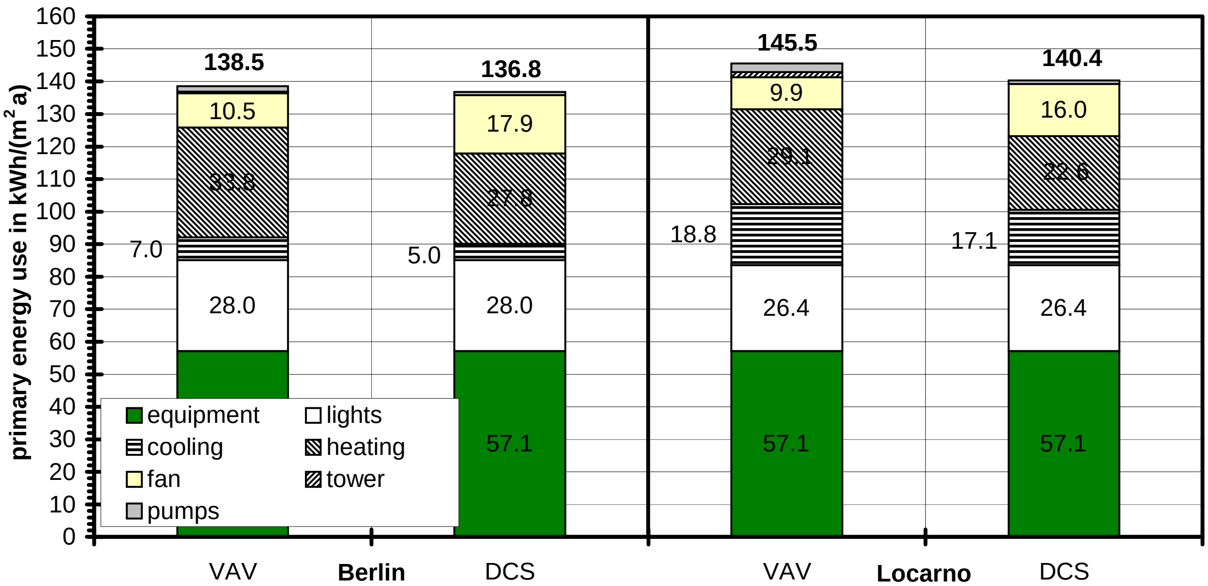

Figure 11.3 : Comparison of primary energy use by category for gas-fired desiccant cooling system (DCS[40]) and the reference VAV system (VAV).

The following Figure 11.4 presents the savings in primary energy in kWh/(m^2 a) by category for the two climates investigated. Figure 11.5 shows the percentage of the savings, always referring to the category's energy use of the reference VAV system. The heating energy consumption of the desiccant cooling system gets reduced as the heat recovery improves with two heat exchangers in series. The primary energy use for cooling decreases with desiccant cooling by 9 % (Locarno) to 28 % (Berlin) when compared with the chiller energy of the reference VAV system. Due to the increased pressure drop of the desiccant cooling system, the

[40] The abbreviation "DEC" for desiccant and evaporative cooling is been used more frequently in Germany.

fan energy consumption exceeds that of the reference VAV system[41]. Obviously, the increased fan energy compensates for the savings within the other categories. The category "HVAC" includes all components for heating and air-conditioning and experiences savings between 3 % (Berlin) and 8 % (Locarno). Although, this is not compelling, it proves that energy efficient air-conditioning without using compressive cooling can provide thermal comfort in these European regions.

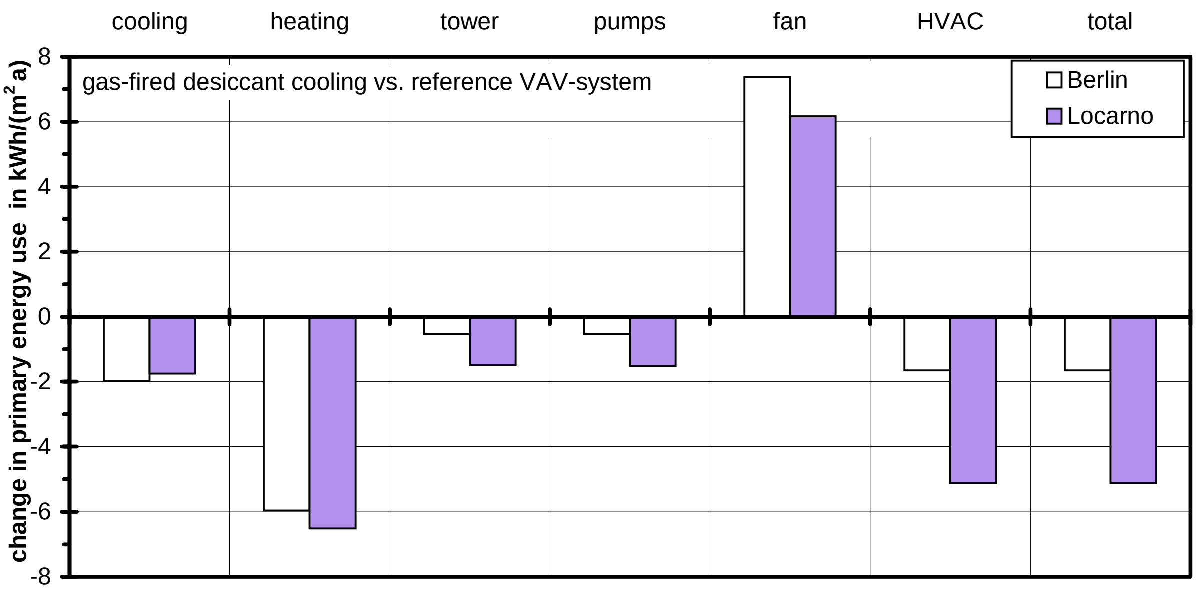

Figure 11.4 : Savings in primary energy by category when a gas-fired desiccant cooling system is used instead of the reference VAV system.

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[41] The pressure drop of the rotary adsorption wheel and the heat exchanger might be improved by selecting bigger rotary wheels resulting in a reduced pressure drop which leads to a reduction of fan power consumption and better system's primary energy use.

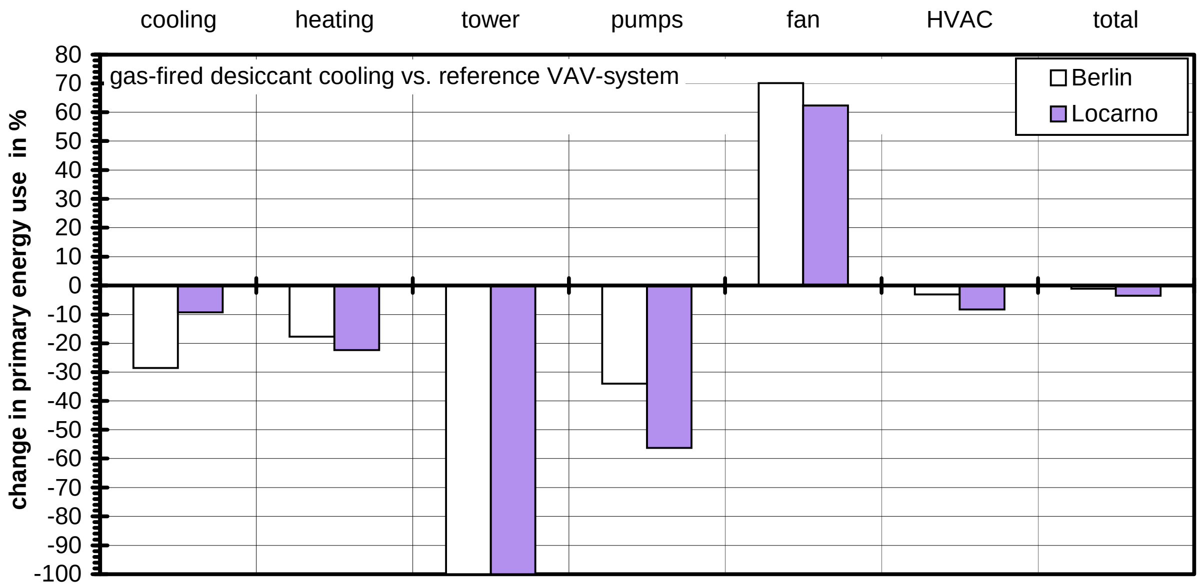

Figure 11.5 : Percentage of savings in primary energy by category when a gas-fired desiccant cooling system is used instead of the reference VAV system.

The highest COP values calculated for gas-fired desiccant cooling occur under part load conditions (almost no dehumidification needed) and are about 1.75 for both Berlin and Locarno. When high humidity ratios require more dehumidification, the COP decreases to 0.6 (Locarno) to 0.7 (Berlin). Considering the whole cooling season, the COP of this system is about 0.95. This is still worse then the performance of modern compression chillers achieving annual COP-values of about 1.0 -1.4. However, desiccant cooling becomes more attractive when the aspect of fuel switching is taken into account. This is reducing the electrical peak power demand and thus, is offering more favorable operating costs for the building.

Table 11.2 shows the changes in electrical peak power demand when desiccant cooling is used in lieu of conventional compressive cooling. For desiccant cooling, the power demand of two different hours is shown for each of the climates. The first hour represents the hour of the highest demand for the reference VAV system. The latter depicts the data for the hour with the highest power demand of the respective desiccant cooling system. Both prove that remarkable reductions in the building's electrical peak power of up to almost 50 % are possible when compressive cooling is replaced by desiccant cooling.

Table 11.2 : Electrical peak power demand in W/m^2 of the reference VAV system and the gas-fired desiccant cooling system.

	Berlin			Locarno		
	VAV	**DCS**		**VAV**	**DCS**	
category	Jul. 9th; 16^{oo} h		Sep. 7th 8^{oo} h	Jul. 23rd; 9^{oo} h		Aug. 24th 9^{oo} h
cooling	12.19	0.29$^{1)}$	0.14$^{1)2)}$	17.74	0.29$^{1)}$	0.29$^{1)}$
tower	0.91	-	-	1.35	-	-
pumps	0.42	0.29$^{1)}$	- $^{2)}$	0.62	0.29$^{1)}$	0.29$^{1)}$
fan	2.37	4.49	4.64	3.54	6.31	4.59
Σ HVAC	15.89	5.06	4.79	23.25	6.88	5.17
peak power reduction		**68 %**	**70 %**		**70 %**	**78 %**
lights	2.67	2.67	4.57	2.36	2.36	5.85
equipment	8.61	8.61	8.43	8.61	8.61	8.61
Σ building	27.18	16.35	17.79	34.22	17.85	19.63
peak power reduction		**40 %**	**35 %**		**48 %**	**43 %**

1) : The electrical power demand for cooling is caused by the motors to rotate the desiccant wheel and the rotary heat exchanger and is set to the estimated electrical input. The power demand of the pump refers to the evaporative coolers (compare Table 11.1).
2) : The humidifiers are not operated at this particular hour.

11.3.2 Regeneration with waste heat

Regenerating the desiccant wheel with a gas-fired heater consumes between 10 % (Berlin) and 30 % (Locarno) of the primary energy of the whole HVAC unit (see Figure 11.3). If this gas consumption could be substituted by another heat source which is either available for free or at least less costly, desiccant cooling would become even more attractive. The most important issue for taking advantage of another heat source is the required temperature level for regenerating the desiccant. The performance of the desiccant wheel depends significantly on the regeneration temperature [2] (compare Chapter 17.6, Appendix), thus, a certain temperature level has to be provided in order to achieve an adequate dehumidification of the outside air.

The following Figure 11.6 presents the cumulative distributions of the regeneration temperature necessary to sufficiently dehumidify the supply air in Berlin and Locarno (results from the gas-fired variation). Moisture needs to be removed from the supply air at about 280 h in Berlin, where regeneration temperatures of up to about 70 ^{o}C are required. The more humid climatic conditions in Locarno demand dehumidification at about 480 h and regeneration temperatures of more than

100 °C. However, much lower temperatures are sufficient to regenerate the adsorption wheel during most hours of desiccant operation. The example in Figure 11.6 shows that desiccant cooling can be performed sufficiently in Locarno with regeneration temperatures of 45 °C or less at about one third of the hours when dehumidification is required. In Berlin, even 62 % of the hours with dehumidification demand can be operated with a regeneration temperature of 45 °C or less.

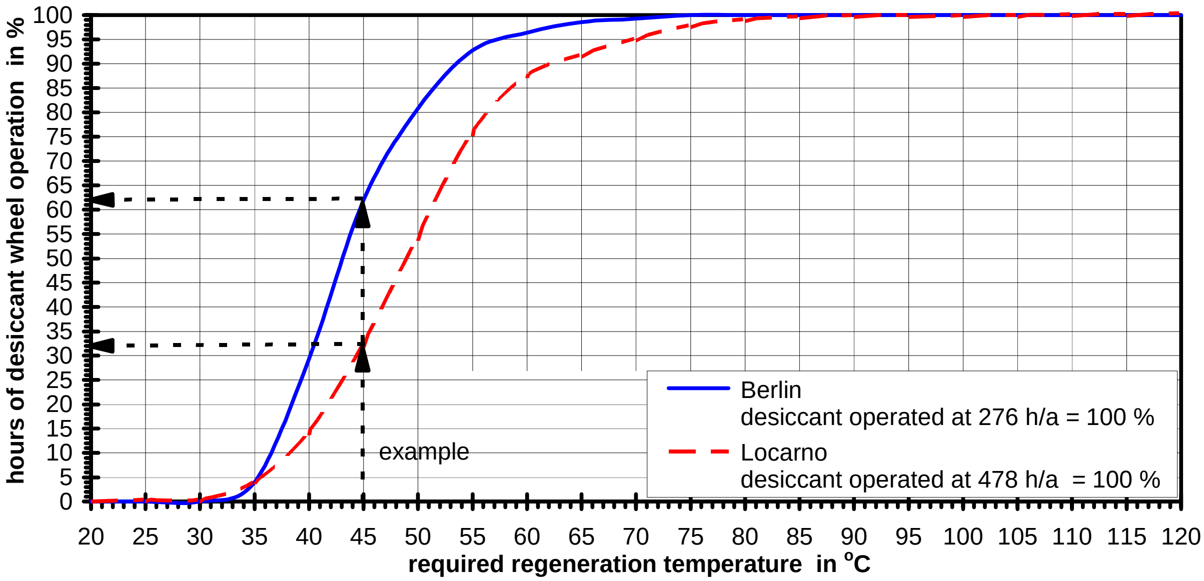

Figure 11.6 : Cumulative distributions of the regeneration temperature required for dehumidification (compare Chapter 17.6, Appendix)

Depending on the temperature level available, waste heat can be used more or less effective instead of natural gas. In this study, the term “waste heat” incorporates waste heat from industrial processes and no primary energy is assumed to generate this heat. Other heat sources like, e.g., district heat, heat from cogeneration plants, fuel cells, solar collectors or geothermal plants, can be used for desiccant cooling as well, but, these sources usually require a certain primary energy consumption. The usage of district heat for desiccant cooling will be presented later.

Based on the results of the gas-fired desiccant cooling variation, the hourly gas consumption is analyzed with respect to the regeneration temperatures. Since the temperature of a waste heat source might not be sufficient to substitute gas entirely, gas and waste heat will be used alternately or in series. This requires an additional heat exchanger slightly increasing the pressure drop for the return air fan.

Figure 11.7 presents schematically the heat provided by gas and waste heat, respectively, operated alternately or in series. The performances of the gas-fired heater and the waste heat exchanger have to be considered when calculating the primary energy use of the system. For this study, a rough estimate for the effectiveness of the waste heat exchanger is made. A temperature differential between the entering waste heat and the leaving regeneration air of 5 K is assumed. It did not seem to be appropriate to use more specified data for the waste heat exchanger, as the boundary conditions might be different for each application.

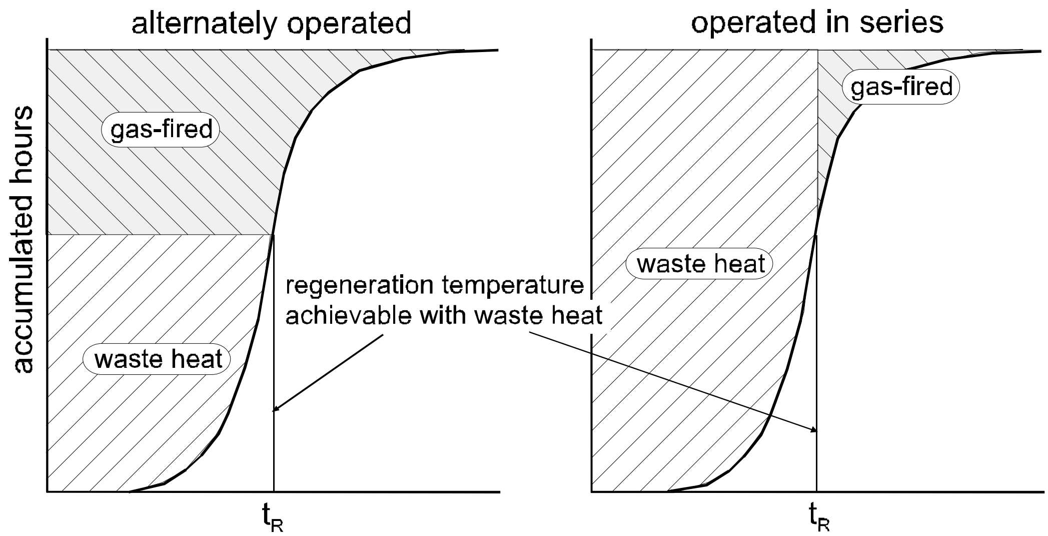

Figure 11.7 : Schematic representation of mutual usage of gas and waste heat

As the temperature level of waste heat sources varies by application, several variations with waste heat usage at different temperatures are investigated. However, it becomes obvious that using waste heat and gas in series is more advantageous than used these alternately. Assuming that waste heat was available at the referred temperature level (compare the legend), the savings for both climates and in-series-operation are presented in Figure 11.8 and Figure 11.9. Only the categories involved are shown graphically[42].

The results shown for a waste heat temperature of 50 °C refer to the example presented in Figure 11.6 as an additional temperature differential of 5 K is considered. A waste heat source with a temperature of 50 °C operated in series with a gas-fired heater, reduces the gas consumption for cooling purposes by 61 % (Berlin) or 36 % (Locarno), respectively, over the only gas-fired desiccant cooling system. The category "HVAC" experiences savings in primary energy for the climate of Berlin of about 5 % over the gas-fired desiccant cooling system when waste heat of 50 °C was available. In Locarno, the gas consumption with additional waste heat usage (50 °C) could be reduced by about 10 %.

[42] If the waste heat was available in wintertime, too, additional savings for heating could be possible, depending on the characteristics of the waste heat. This additional advantage is not taken into account in this study.

Comparing the waste heat-operated desiccant cooling system, using waste heat with a temperature of 50°C, with the reference VAV system with compression chiller, the primary energy use for cooling can be reduced by about 42 % (Locarno) to 72 % (Berlin). Savings in system's primary energy consumption of about 8 % (Berlin) to 18 % (Locarno) can be achieved.

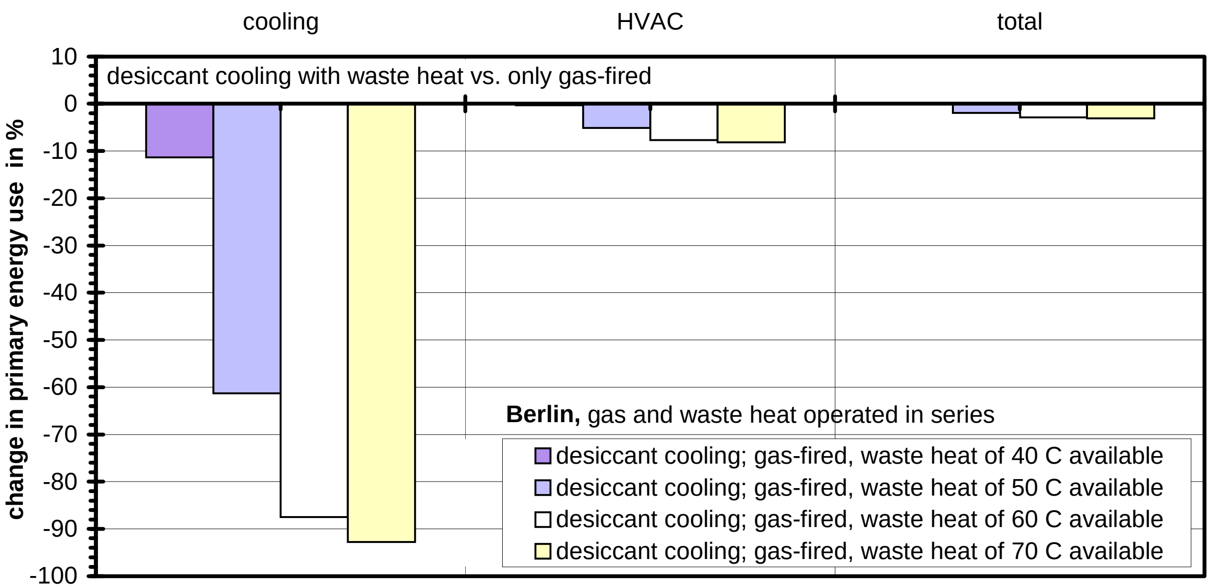

Figure 11.8 : Savings in primary energy over the only gas-fired desiccant cooling system when gas and waste heat are used in series in Berlin

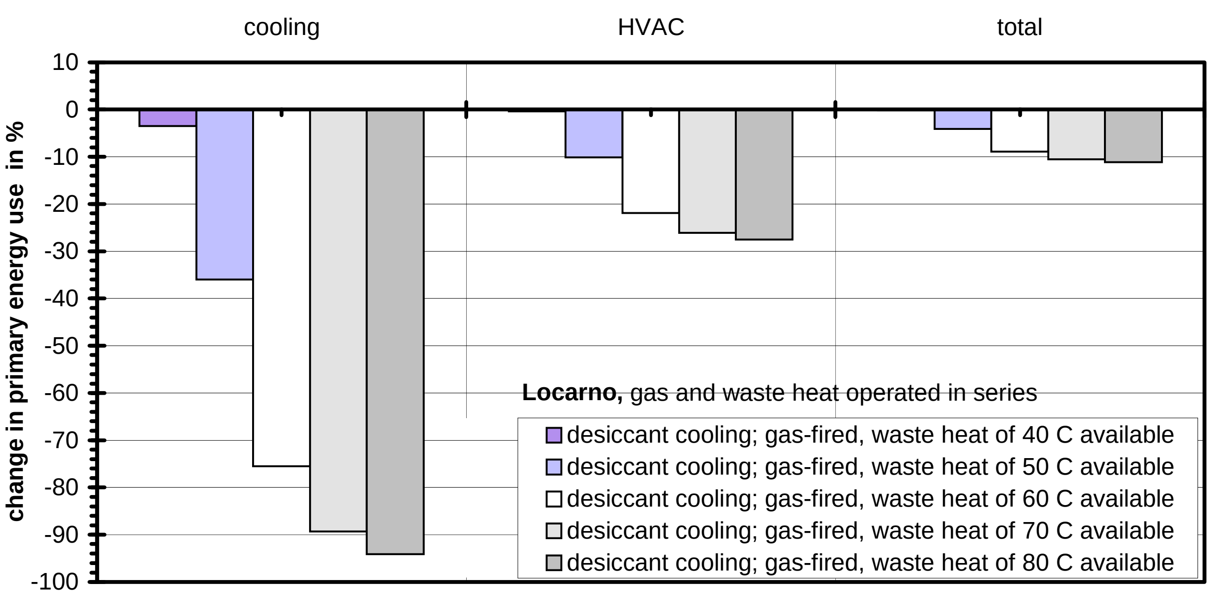

Figure 11.9 : Savings in primary energy over the only gas-fired desiccant cooling system when gas and waste heat are used in series in Locarno

11.3.3 Regeneration with district heat

Unfortunately, waste heat from industrial plants is not available everywhere. In Germany for example, about 75 % of the district heat is made in electrical power plants. The district heat distribution covers about 9 % of all residential and non-residential buildings in the former West Germany and about 25 % in the former East Germany. The portion of buildings heated that way is still increasing, especially in the downtown areas of the bigger German cities [34]. In these areas, district heat could be used for cooling purposes in summertime, too. During summertime, the supply water temperature of district heating systems is usually between 65 ^{o}C and 85 ^{o}C (Germany). In general, plenty of district heat is available in summertime, so using this heat source would result in a better efficiency of the power plants and would reduce the production of CO_2.

Attractive rates for this energy source should be offered by the utlities, as they ought to be interested to optimize their plants, in terms of primary energy efficiency. However, the current prices for district heat both in Germany and the United States do not confirm this.

The electrical effectiveness of a power plant decreases slightly when heat is been generated in addition to the electricity [34]. Figure 11.10 shows common numbers for the performance of a power plant featuring co-generation.

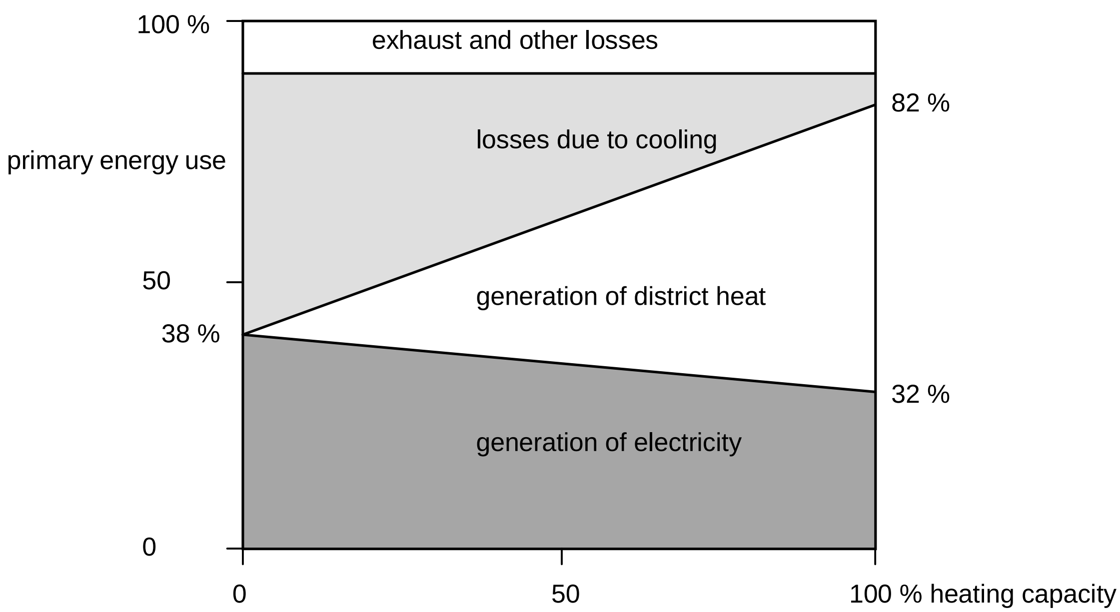

Figure 11.10 : Characteristics of a power plant generating both electricity and district heat

Taking into account the numbers presented in Figure 11.10 and additional heat losses of 20 % between the power plant and the customers, about 0.38 kWh primary energy are consumed to provide 1 kWh district heat to the desiccant cooling system.

The temperature level of district heating systems in Germany is usually sufficient to provide satisfying regeneration temperatures throughout the cooling (dehumidification) season. Therefore, the following results for Berlin presented in Figure 11.11 assume only district heat as heat source. In Locarno, higher regeneration temperatures are required, thus, district heat and gas are been used in series. In this case, a district heat temperature of 70°C is assumed.

When district heat (70 °C) can be utilized to operate a desiccant cooling system, the primary energy use for cooling or dehumidification, respectively, decreases by about 62 % (Locarno) to 72 % (Berlin) when compared with the reference VAV system. Referring to the HVAC system, savings in primary energy of about 8 % (Berlin) to 25 % (Locarno) are possible, when district heat is been used to regenerate the desiccant.

As district heat is not for free, the energy costs have to be taken into consideration, when this strategy is evaluated. Based on a market survey in Germany [29] and a study about commercial buildings in the U.S. [55] rates for district heat are used to determine the annual energy costs. A comparison between the rates for district heat and natural gas shows no significant benefits from using district heat instead of gas. However, financial savings over the gas-fired variation occur as less primary energy is consumed with district heat. The annual energy costs are shown and evaluated in Chapter 14.

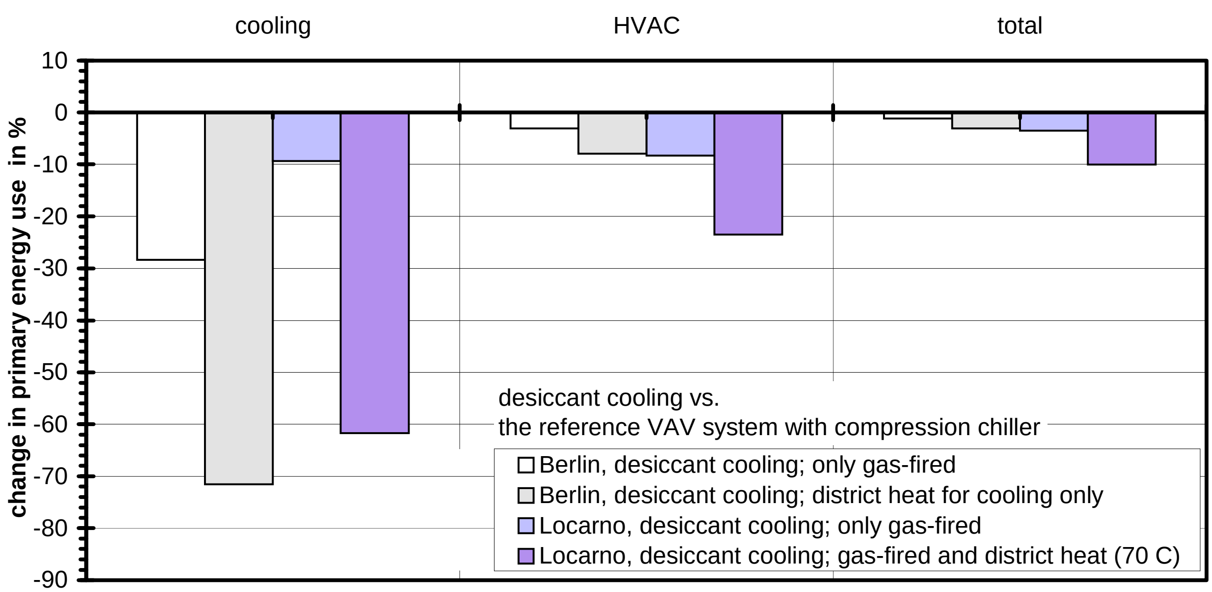

Figure 11.11 : Savings in primary energy over the reference VAV system when a desiccant cooling system is been operated with gas and district heat

Additional savings in primary energy can be achieved when district heat is utilized for heating purposes in wintertime. Although, heat losses from the power plant to the customer have to be taken into account, the primary energy effort to provide heat is rather low. Assuming heat losses of about 5 % [34], approximately 0.33 kWh primary energy is consumed to provide 1 kWh of district heat to the customer. However, the heating energy consumption of the building under study is not a major issue of this investigation.

11.3.4 Summary

Desiccant cooling is able to compete with compressive cooling in terms of primary energy consumption. Depending on the climatic conditions, the primary energy used for cooling purposes, i.e., for regenerating the desiccant , will be lower when compared with conventional HVAC systems using compressor-driven chillers. Due to increased pressure drops, the fans consume more energy with desiccant cooling than with the reference VAV system. The main benefit for a gas-fired system, however, arises from the enhanced heat recovery as two rotary heat exchangers can be used to recover heat during wintertime. Using gas for regeneration reduces the primary energy consumption of the building slightly, but taking fuel switching into account favors desiccant cooling significantly. The electrical peak power demand of the building can be reduced by about 35 % (Berlin) to 43 % (Locarno) when gas-fired desiccant cooling is used in lieu of compressive cooling.

However, the most favorable boundary condition for desiccant cooling constitutes a waste heat source available for free with a suitable temperature level for regeneration. Most likely, gas and waste heat would be used in combination, preferably in series. This requires an additional heat exchanger, which slightly increases the pressure drop of the return air system. Depending on the temperature level of the waste heat source, the gas consumption can be reduced remarkably when compared with only gas-fired desiccant cooling system. The primary energy consumption of the HVAC system taking advantage of waste heat can be reduced to up to 11 % (Berlin) or even 32 % (Locarno) when compared with the reference VAV system. When waste heat is not available, e.g., district heat can be used instead.

District heat from power plants appears to be an interesting heat source for desiccant cooling in regions where district heat is been used for heating in winter anyway. Better utilization of the fuels in summer increases the effectiveness of the power plant and reduces its CO_2-production. The primary energy use for cooling, respectively for dehumidification, can be reduced significantly using district heat for desiccant cooling. Operating desiccant cooling with district heat can reduce the primary energy use of the HVAC system by about 8 % (Berlin) to 25 % (Locarno).

Considering the savings possible with waste heat usage, taking advantage of solar energy seems to be favorable. Although, the current price situation for solar collectors might not offer realistic payback times, this combination of energy sources could become more attractive in the near future.

12. Absorption Chiller

The cooling capacity to cool and dehumidify the supply air sufficiently can also be provided by an absorption chiller instead of a compression chiller. In both cases, taking up heat at a low pressure evaporates a liquid refrigerant. Then the pressure of the refrigerant is elevated and the heat is rejected to the ambient converting the vaporized refrigerant back into liquid. The major difference is the energy source used for increasing the pressure of the refrigerant. The compression chiller uses mechanical energy, commonly provided by electricity. Absorption chillers are mainly using heat instead, e.g., steam, hot water. The latter consumes electrical energy only to operate a solution pump using about 10 % of the electricity consumed by a compression chiller, which also decreases the electrical peak power demand. Unfortunately, the absorption process needs to reject a higher amount of heat requiring a bigger cooling tower and consuming more water. Absorption chillers are usually more expensive than compression chillers but need less maintenance. The absorption process is the only alternative cooling strategy investigated which both can substitute the compression chiller entirely and is able to provide chilled water at the same temperatures.

An clear advantage over the compression cooling process is offered by the refrigerants used in the absorption cooling process which are less harmful to the environment than CFCs, HCFCs or HFCs. The refrigerants used in the heat-operated absorption process for air-conditioning purposes are usually water (in combination with lithium bromide) or ammonia (with water as absorbent). Figure 12.1 presents a LiBr-H_2O absorption process by its general components.

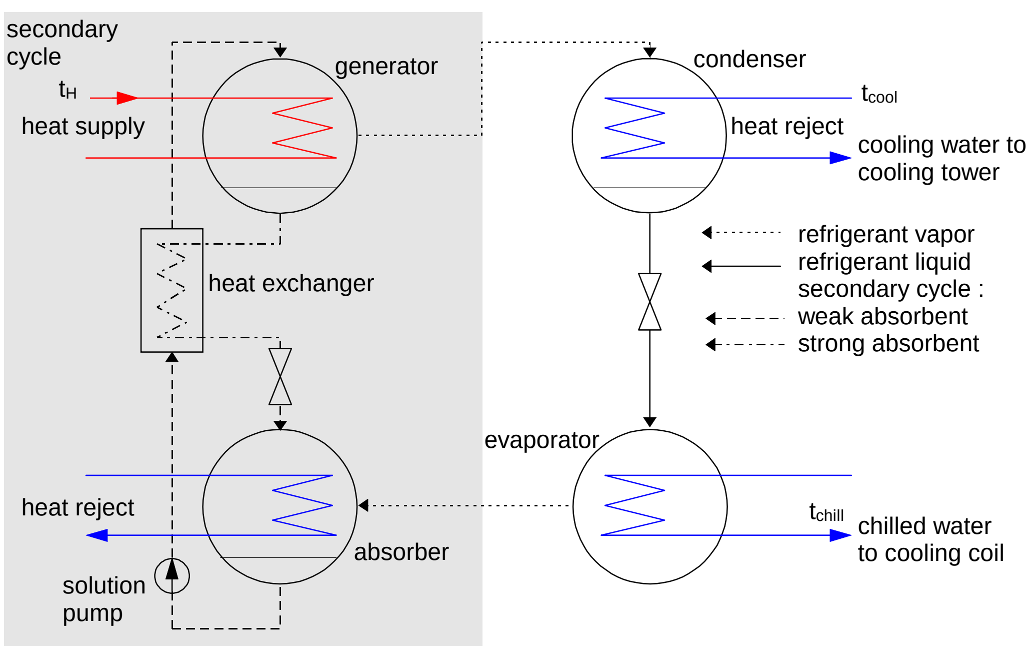

Figure 12.1: Schematic of a single-stage LiBr-H_2O absorption chiller

Removing heat from the chilled water in the cooling coil cycle (evaporator) evaporates the liquid refrigerant. In the absorber (secondary cycle), the water vapor is been absorbed by a lithium bromide (LiBr) solution rejecting absorption heat to the cooling water. Then the solution pump delivering the liquid to the generator elevates the pressure of the weak absorbent. Heat supplied to the generator separates the refrigerant and the absorbent by evaporating the water, which is condensed in the condenser rejecting heat to the cooling water running through the cooling tower. Then the refrigerant is expanded and transferred to the evaporator as a liquid. The remaining high pressure liquid in the generator (strong absorbent) is used to pre-heat the weak absorbent (heat exchanger) to increase the efficiency of the process and is then expanded before it returns to the absorber.

Single-stage absorption chillers using $LiBr-H_2O$ require heat supplied at temperatures of about $t_H \approx 70 - 100\ ^oC$ and normally achieve COP-values of about 0.65 - 0.75[43]. When waste heat with a fairly high temperature level is available at low costs or even for free, utilizing single-stage absorption chillers can become favorable even with the relatively low efficiency. Two-stage or dual-effect machines perform much better ($COP_{two\text{-}stage} \sim 1.0 - 1.35$) but also demand higher temperatures ($t_H \approx 110 - 150\ ^oC$) at the two generators [24]. These chillers are commonly gas-fired, unless high pressure hot water or steam is available. Considering these COP-values, it becomes clear that the absorption process demands about the same or even more primary energy as a compression process.

12.1 Gas-fired absorption chiller

Both single-stage and dual-effect absorption processes with the generators heated by gas are compared in this section. The following results are based on design-COP values of 0.70 for the single-stage process and 1.20 for the dual-effect absorption chiller. These COPs refer to heat supplied at about 70 ^oC (single-stage) and 130 ^oC (dual-effect), respectively [24]. Under part load conditions the performance of the chiller is adjusted to the part load ratio. The design-COPs are achieved with a supply temperature of the chilled water of about 7 ^oC and a cooled water temperature (from the cooling tower) of about 29 ^oC. The effectiveness of the gas-fired heat exchanger was set to 0.90.

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[43] The COP of an absorption chiller is defined as ratio between cooling energy provided to the heating energy input. The electrical energy to run the soultion pump can usually be neglected. In contrast with that, the performance of a compression chiller refers to the electrical input, thus, when the performance of an absorption chiller shall be compared with a compression chiller, the primary energy input should be used in both cases.

At design conditions, highly developed compression chillers usually offer a ratio between cooling energy provided and electrical energy input of about 3.5 to 4.5. Considering the conversion losses when electricity is generated, primary energy based COP-values of about 1.2 to 1.5 are reached. However, this study is calculating the energy use of the compression chiller with a design-COP (primary energy based) of 1.06. Better performance will be considered in Chapter 14

12.1.1 Primary energy consumption

The following Table 12.1 summarizes the coefficients of performance of the two different gas-fired absorption chiller processes and compares these with those of the compressive cooling investigated (reference VAV system). The values shown are based on the chiller's primary energy input and cooling energy output and refer to the entire cooling season, i.e., these are annual performance values.

Table 12.1 : Comparison of performance of the compression chiller, single-stage and dual-effect absorption chiller (see also Figure 12.2)

category		location				
		Berlin	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
COP_{chill} 2)	compression chiller 1)	0.99	0.98	1.00	1.02	0.98
	single-stage	0.59	0.58	0.61	0.65	0.56
	dual-effect	0.94	0.93	0.97	1.05	0.89

1) The electric input ratio of the compression chiller is set to 0.33
2) Considers primary energy including the electrical conversion and the entire cooling period.

The annual primary energy use of both the single-stage and the dual-effect absorption chillers are compared in Table 12.2, referring to the primary energy use of the compression chiller system. The data show that the cooling requirement influences the system's efficiency. In moderate climates, the primary energy use of compression chillers is apparently more favorable than the one of the single-stage and even the dual-effect gas-fired absorption chillers, as the annual COP values of the compression chiller are higher (compare Table 12.1). The best performance of each of the cooling systems is provided with the climatic data of Red Bluff, which has the highest cooling demand of the climates under study. In this particular climate, the dual-effect absorption chiller performs even better than the compression chiller with an electric input ratio of 0.33. However, in all the other climates investigated, both absorption processes require more primary energy for cooling than conventional cooling. Due to the increased energy use of the cooling tower, even in Red Bluff the primary energy consumption of the HVAC system with a dual-effect absorption chiller is about 3 % higher than that of a compressive cooling system (Table 12.2, Figure 12.3).

If highly developed compression chiller offering better performance, e.g., electric input ratio = 0.25, were being used for the reference system, the absorption process would even come off worse.

Table 12.2 : Primary energy use of gas-fired absorption chillers in % compared with that of a compression chiller (= 100 %); (see Figure 12.3)

category		location				
		Berlin	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
single-stage process COP = 0.70	$Q_{a,chill}$ [%]	169	172	164	157	176
	$Q_{a,HVAC}$ [%]	**111**	**108**	**123**	**132**	**121**
	$Q_{a,total}$ [%]	104	103	110	105	116
dual-effect process COP = 1.20	$Q_{a,chill}$ [%]	106	108	103	97	111
	$Q_{a,HVAC}$ [%]	**102**	**102**	**104**	**103**	**106**
	$Q_{a,total}$ [%]	101	101	102	101	101

The change in primary energy use with the single-stage absorption process is shown in the following Figure 12.2 and the results with the dual-effect absorption chiller are presented in Figure 12.3. The differences in kWh/a are only shown by categories involved. In both cases, the results with the reference VAV system using compressive cooling constitute the basis for the comparison.

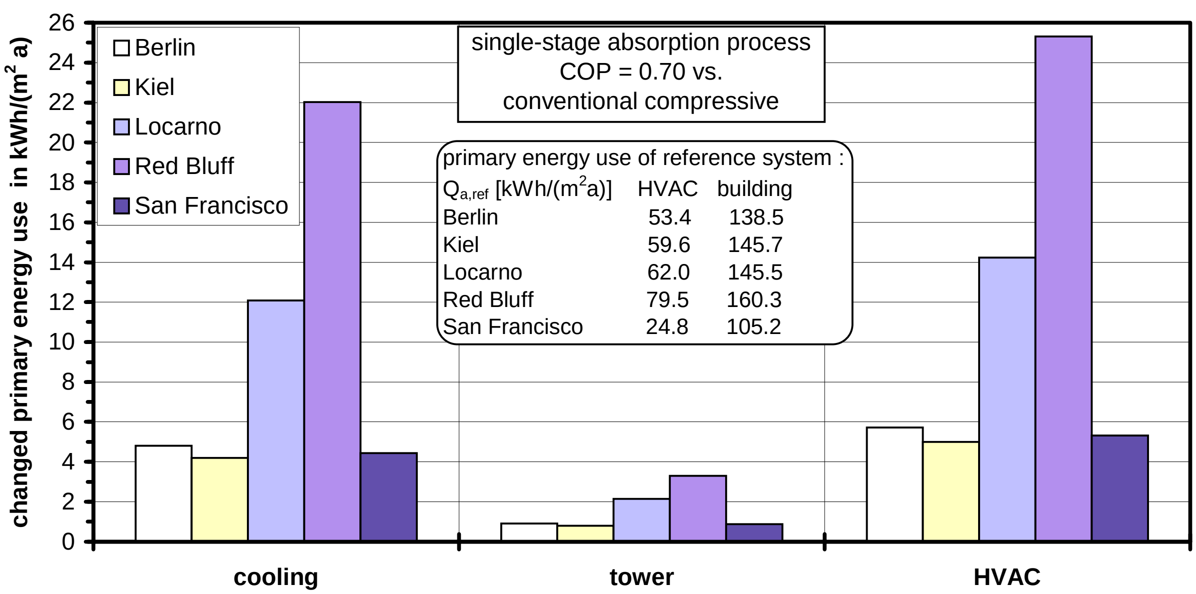

Figure 12.2 : Change in primary energy use in kWh/a operating a single-stage absorption chiller instead of a compression chiller; compare Table 12.2

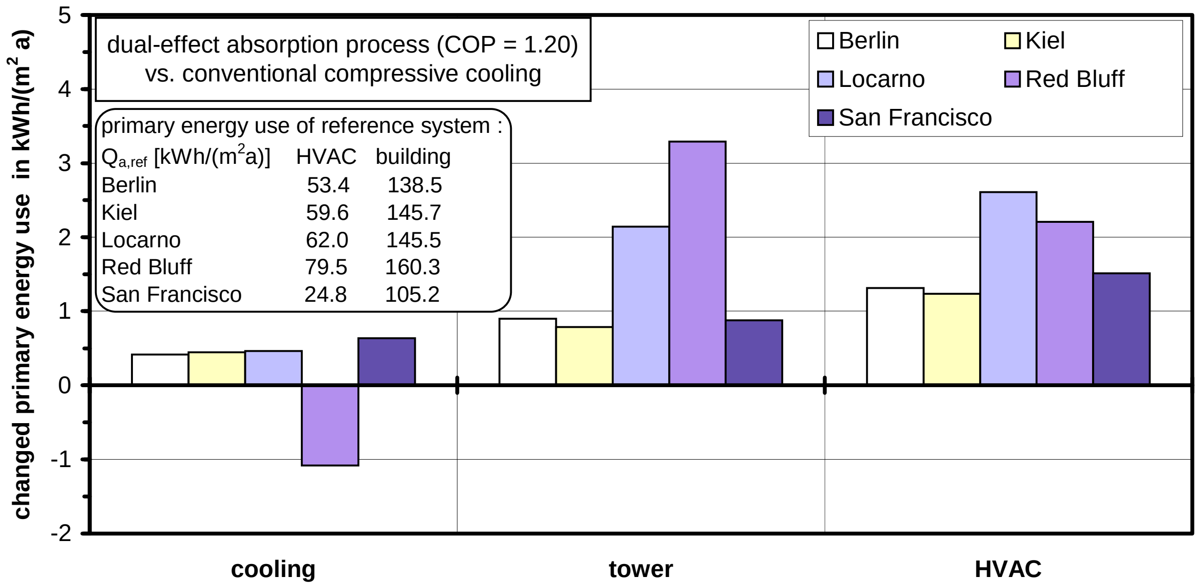

Figure 12.3 : Change in primary energy use in kWh/a operating a dual-effect absorption chiller instead of a compression chiller; compare Table 12.2.

Considering the primary energy consumption of the building utilizing gas-fired absorption chillers to provide the cooling energy, appears to be no competitive method. The primary energy use of the respective conventional system with compression chiller outperforms the gas-fired single-stage absorption chiller clearly and is still slightly better than that of the gas-fired dual-effect absorption chiller.

12.1.2 Electrical power demand

However, not only the primary energy consumption is an important issues when evaluating different cooling strategies but the electrical peak power demand as well. Using an absorption chiller instead of a compression chiller, reduces the electrical power demand significantly, resulting in more favorable electricity rates. With absorption chiller operation, the peak power demand is determined by the power demand of the lighting and the equipment rather than by the cooling demand.

Figure 12.4 compares the building's electrical peak power demand by category at the respective peak hour. The peak power reductions apply to both the single-stage and the dual-effect absorption chiller since these do not differ significantly. As the solution pump of the absorption process demands only about 5 % of a compressor with the same cooling capacity the peak power reduction possible is particularly high when the cooling energy requirements are high, too (compare Figure 12.4; Locarno and Red Bluff). The building's peak power demand including lights and electrical equipment can be reduced by 23 % (San Francisco) to 48 % (Red Bluff). Considering only the HVAC system's energy use, the peak power demand can be reduced by 66 % (Red Bluff) to almost 89 % (Berlin; see Table 12.3).

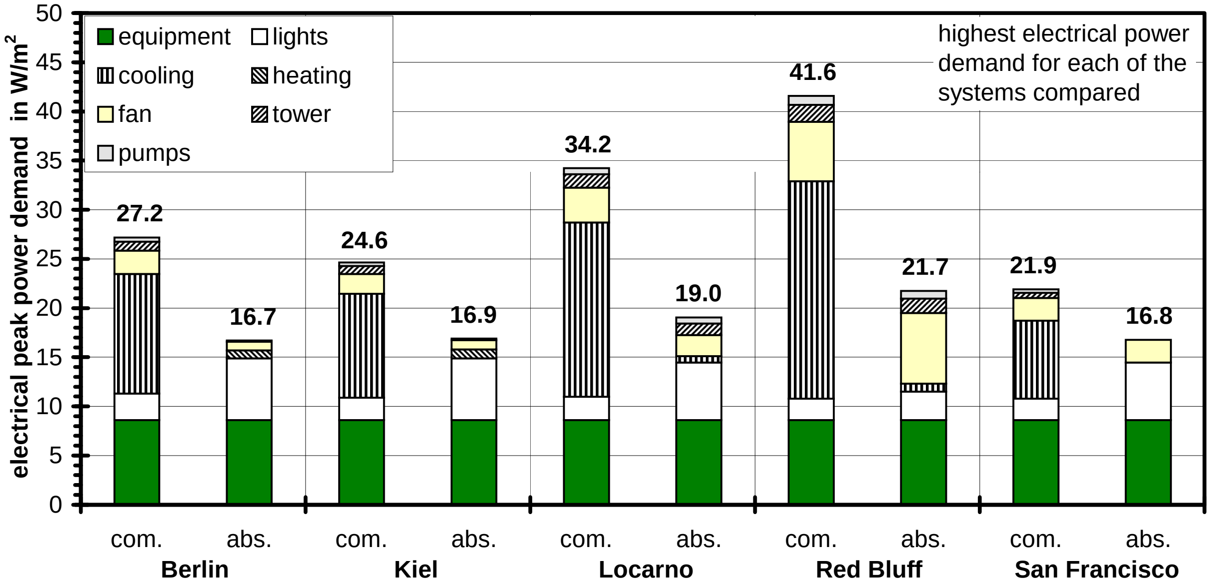

Figure 12.4 : Electrical peak power reduction in W/m^2 when an absorption chiller (abs.) is used instead of a compression chiller (com); compare Table 12.3.

Table 12.3 : Peak power demand of a single-stage and a dual-effect absorption chiller in %, compared with the reference VAV system with compression chiller (= 100 %; see also: Figure 12.4)

	location				
category	**Berlin**	**Kiel**	**Locarno**	**Red Bluff**	**San Francisco**
HVAC in % 1)	11	15	20	33	21
entire building in % 2)	61	69	56	52	77

1) The category HVAC contains the energy use of cooling, heating, pumps, tower and fans.
2) The building's power demand includes all categories, i.e., lights and equipment, too. The power demand is related to the power demand of compressive cooling and almost the same for both the single-stage and dual-effect absorption cooling.

12.1.3 Waste heat usage

If waste heat were available at a temperature of about 70 - 90 °C, using a single-stage absorption chiller instead a compression chiller would result in a reduction of primary energy consumption despite the worse COP value. Just to demonstrate the saving potential, Figure 12.5 compares the data for the single-stage absorption process, assuming that no gas consumption for cooling purposes was required, with the reference VAV system. Primary energy savings for the HVAC system between 7 % (Kiel) and 40 % (Red Bluff) could be possible.

In areas where district heat is been produced in electrical power plants, i.e., as a by-product of the electricity generated, district heat appears to be a suitable heat source for the absorption process. The temperature level of the warm water provided can exceed 100 °C even in summertime, but might be as low as 70 °C. In Germany, the usual temperature of district heat is between 65 °C to 85 °C. Due to these temperatures, only single-stage absorption chillers are eligible in conjunction with district heat.

The savings potential for taking advantage of district heat in single-stage absorption chillers for the German climates reach almost 90 % of the cooling energy or between 7 % (Kiel) and 9 % (Berlin) of the HVAC energy when compared with conventional compressive cooling (reference VAV system). Considering the potential to reduce the electrical power peak (see Figure 12.4 and Table 12.3), using an single-stage absorption chiller with district heat becomes even more attractive.

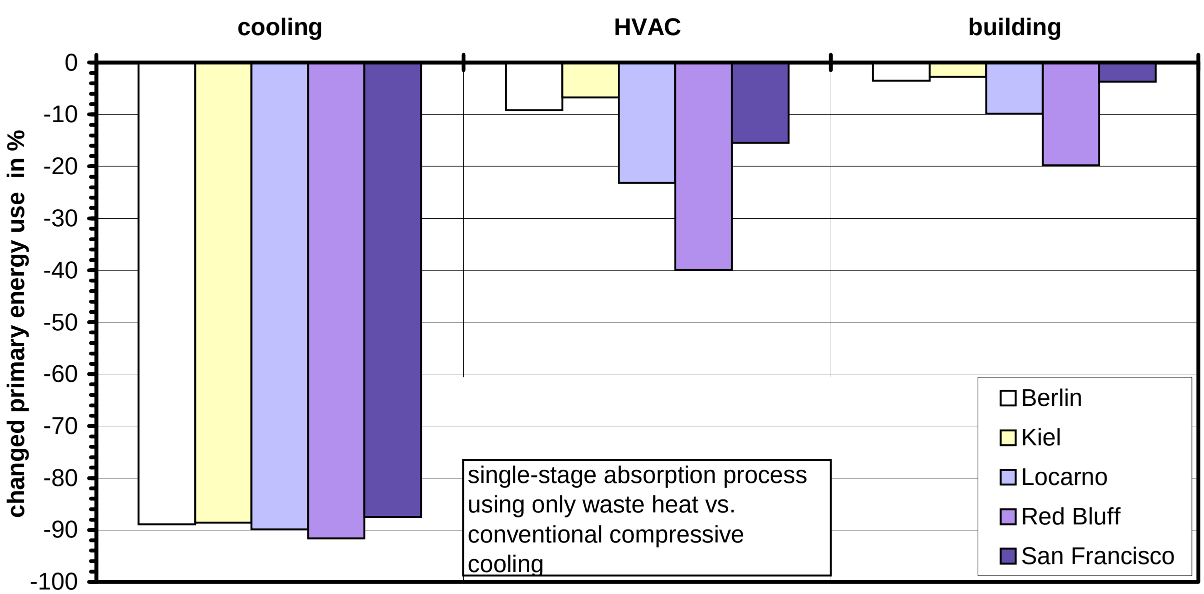

Figure 12.5 : Primary energy savings of a single-stage absorption chiller being operated with waste heat only over the reference VAV system (compression chiller)

12.2 Solar-assisted absorption chiller

Based on the results presented above, solar-assisted absorption chiller operation appears to be a smart solution for climates with high solar radiation. “Solar-assisted” refers to absorption chillers, which are heated by warm water from solar collectors. If the heat collected is not sufficient to heat the generator, gas is fired additionally.

A study about operating an absorption chiller in combination with solar collectors was conducted at LBL and was published in 1997 [47]. Climatic data for the Californian regions San Francisco, Red Bluff and Palm Springs were taken into consideration. Building and system simulation runs, with a single-stage absorption chiller heated by gas or warm water from solar collectors, were performed with DOE-2E and TRNSYS. The office building used for that study corresponds to the building described in Chapter 7, but as the building and the HVAC system were not optimized in the same way, slightly different cooling loads and energy use occurred. However, the savings in primary energy and operation costs possible illustrate the potentials of this system clearly. All results presented in the following Chapter 12.2.3 originate from the investigation carried out by Stetiu and Feustel [47].

12.2.1 Solar collectors

Solar collectors[44] are used to heat water by absorbing solar radiation. In combination with circulation pumps, controls and a storage tank, solar collectors are suitable to produce domestic hot water or even to heat buildings. The performance of a solar heating system, however, depends highly on the solar energy available, the outside air temperature, the required hot water temperature and the construction of the collector itself.

Solar collectors for heating purposes are commonly flat-plate types and are fixed in their orientation and tilt. Figure 12.6 shows a cross-section of a typical solar collector for heating purposes. The absorber plate collects the solar radiation and usually consists of coated copper with attached or integrated pipes. The heat absorbed is being transferred to a water cycle heating water in a hot water storage tank (Figure 12.7). Usually glycol is added to the water flowing through the collector’s cycle to prevent freezing. The glazing on top of the collector is needed to decrease the heat transfer from the absorber plate to the ambient which reduces the heat losses and thus, increases the collector’s efficiency. Furthermore, utilizing transparent insulation materials (TIM) to cover the collector instead of glass is able to reduce the heat losses even better, resulting in very efficient solar collectors [61].

[44] Basics of solar collectors and corresponding components can be found for example in the ASHRAE handbook “HVAC Systems and Equipment” [2]

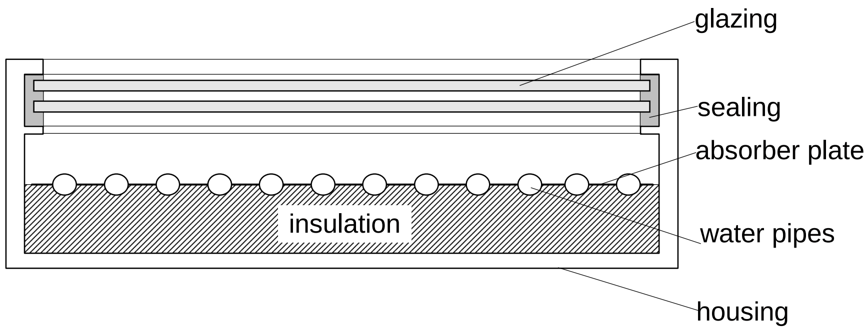

Figure 12.6 : Schematic of a flat-plate solar collector with double glazing

12.2.2 Model

As preliminary simulation results showed that water temperatures of about 85 - 90 ^{o}C can be provided by the solar collectors at best, a single-stage absorption chiller with a design COP-value of 0.70 was used (compare Chapter 12). To calculate the performance of the absorption chiller, the TRNSYS-subroutine “TYPE 7” was modified. The characteristic curves of the chiller within this model were adjusted to manufacturer’s data, the control strategy was refined and a chilled water storage tank was included to let the absorption chiller run more often at design conditions. Figure 12.7 presents the solar-assisted absorption chiller system which was entirely modeled and calculated with TRNSYS. The building’s loads and the cooling requirements, i.e., the required chiller capacity, however, were determined before using DOE-2E.

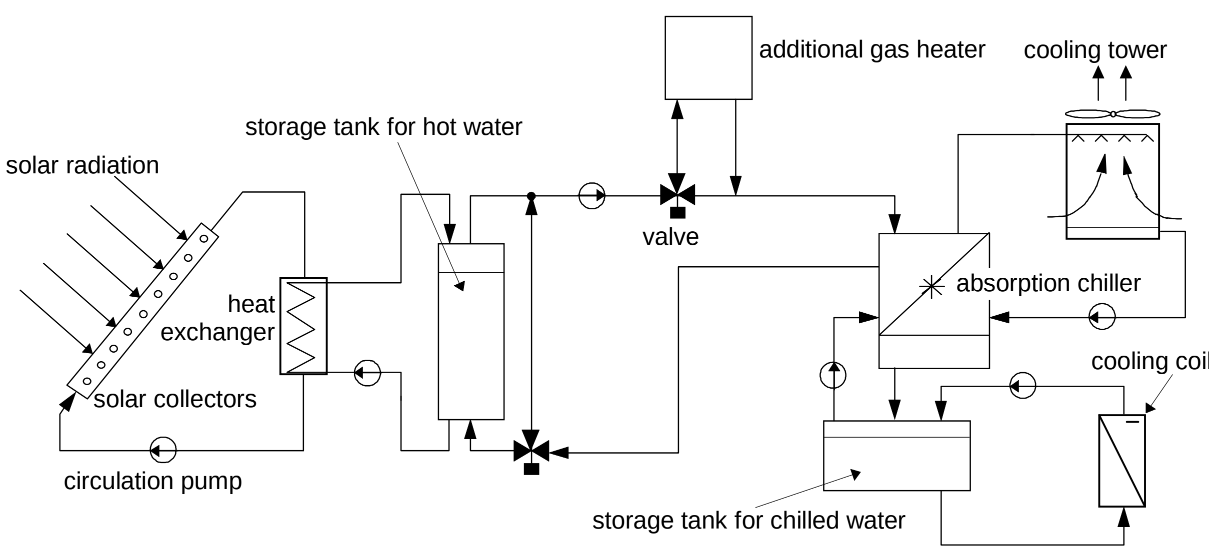

Figure 12.7 : Schematic of the solar-assisted absorption chiller system modeled in [47]

12.2.3 Results

The goal of the referring study [47] was to point out the cooling power potential of solar-assisted absorption chillers and to evaluate the capability to economically compete with conventional compression chillers or sole gas-fired absorption chillers. Consequently, both the life-cycle costs and the thermal performance were decisive for evaluating the cooling system.

First, the size of the hot water storage tank was varied in order to find out the set-up with the most favorable additional auxiliary gas consumption. Tank volumes of 14 l per m^2-collector area (San Francisco) or 20 l/m^2 (Red Bluff, Palm Springs) were chosen. Then, the auxiliary gas consumption of the system was determined with different areas of solar collectors.

The auxiliary primary energy required for heating the absorption chiller process as a function of the solar collector area installed, is presented in Figure 12.8. Curves for two types of solar collectors are shown applying to the Californian climates of Red Bluff and San Francisco. A collector area of 0 m^2 represents an only-gas-fired single-stage absorption chiller. Due to the slightly different building details and system set-ups, the energy consumption shown in Figure 12.8 may not be compared directly with the values shown in the preceding Chapter 12.1. The example in Figure 12.8 shows possible reductions in auxiliary gas consumption between 52 % and 66 % when the chiller is heated by gas and additionally by the double-glazed solar collectors having a size of 50 m^2. It becomes obvious that the collectors covered with a transparent insulation (TIM) outperform the collectors with double-glazing clearly, however, these are more expensive.

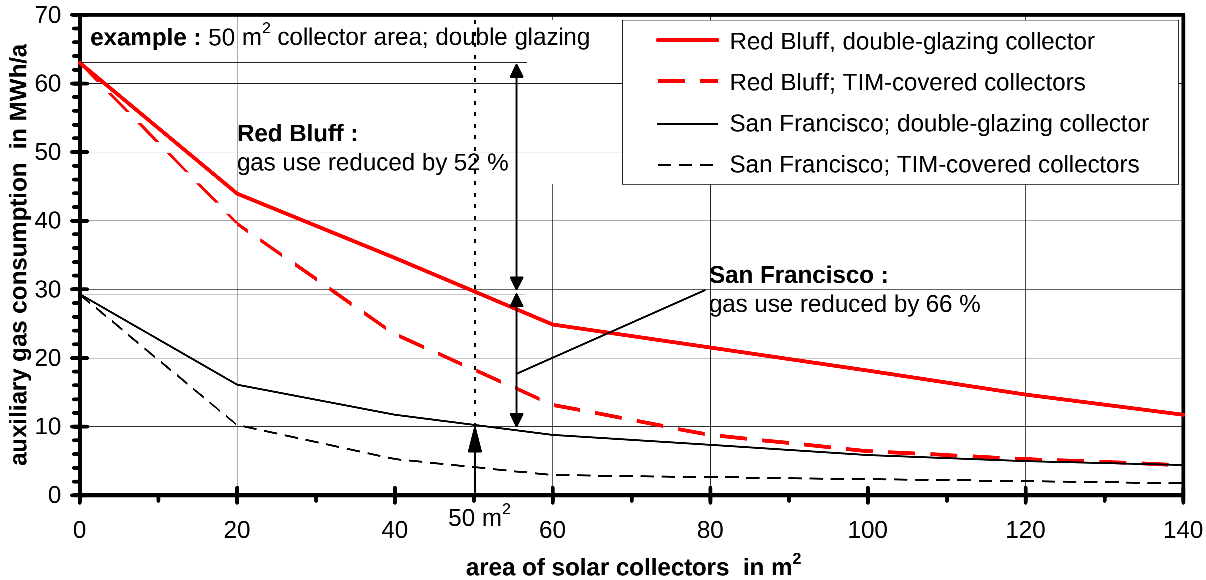

Figure 12.8 : Auxiliary annual gas consumption for different types of solar collectors and the climates of Red Bluff and San Francisco as a function of the collector area [47].

Based on the first costs, interest rates and energy expenses of a gas-fired single-stage absorption chiller system, the most favorable collector area for the solar-assisted system for each of the climates was determined. A survey of the current market in the U.S. led to the following prizes considered in the life-cycle cost-analysis :

equipment :	- reciprocating compression chiller	: 430 US$/kW
	- single-stage absorption chiller	: 330 US$/kW
	- solar collectors; double glazing	: 150 US$/m^2
	- solar collectors with TIM	: 180 US$/m^2
	- hot water storage tank	: 1800 US$/m^2
	- miscellaneous	: 40 US$/m^2
operation :	- electricity	: 0.11 US$/kWh
	- gas	: 0.02 US$/kWh

A system's lifetime of 20 years, annual interest rates of 11 % and an escalation of fuel prices of about 5 % over twenty years were taken into account, too.

Considering the installation and operation costs as well as interest rates and the lifetime, annual costs for each of the systems and climates were calculated. Table 12.4 shows the most cost-effective collector area for the solar-assisted absorption chiller system. If conventional, double-glazed solar collectors are used in the coastal regions of California (San Francisco), no financial benefits over the gas-fired absorption chiller system are possible. The more efficient TIM-covered collectors, however, offer saving potentials in this climate. The higher solar radiation values and the higher cooling requirements in the other two climates favor the solar-assisted absorption chiller over the gas-fired for both of the solar collector types. In these cases, best life-cycle costs can be achieved with solar collector arrays of 20 m^2.

Table 12.4 : Cost-effective solar collector area for different Californian climates [47]

	most efficient solar collector area in m^2	
climates investigated	double-glazed collector	TIM-covered collector
San Francisco (coastal region)	not economic	10
Red Bluff (Central Valley)	20	20
Palm Springs (desert)	20	20

Based on the differences in first costs and the savings in energy, simple payback times for the solar-assisted absorption chiller system were determined using both the compression chiller system and the gas-fired absorption chiller system as reference. These results presented in Table 12.5 do not consider interest rates.

The additional installation of solar-collectors in the coastal regions of California and the Central Valley results in payback times below 10 years when compared with the gas-fired absorption chiller system. With the desert climate of Palm Springs payoff can be expected within 6 or 7 years, respectively. However, if the solar-assisted absorption chiller system is compared with the compression chiller system, the payback times become much more attractive. Payoff within 4 years or even 2 years is possible in the hot and sunny Californian climates of Red Bluff and Palm Springs. The temperate climate of San Francisco requires about three years of operation to achieve energy savings matching the higher expenses of the solar-assisted absorption chiller system.

These payback times are within the common lifetime of an absorption chiller system, i.e., the savings in energy are able to compensate for the higher first costs for the additional solar-collectors. However, usually payback times less than five or even three years are considered to represent an attractive (financial !), alternative energy source.

Table 12.5 : Simple payback time for the solar collectors of the solar-assisted absorption chiller system in years for the Californian climates investigated

climates	base of the comparison	simple payback time in years double-glazed collectors	simple payback time in years TIM-covered collectors
San Francisco	gas-fired absorption	--	9
	compression	--	3
Red Bluff	gas-fired absorption	9	8
	compression	4	4
Palm Springs	gas-fired absorption	7	6
	compression	2	2

In any case under study, single-stage absorption chillers can be heated effectively by solar collectors. Making use of solar energy for cooling purposes, reduces the primary energy consumption by about 21 % (Palm Springs) to 33 % (San Francisco) when compared with an only-gas-fired single-stage absorption chiller. The electrical energy use of the solar-assisted absorption chiller system decreases to less than 15 % of a comparable compressive cooling system and the electrical power demand of the HVAC system at peak cooling load conditions amounts to less than 10 %. However, the solar-assisted single-stage absorption chiller still consumes between 15 % and 31 % more primary energy than the compression chiller of the reference VAV system.

13. Cooled Ceiling

Basic information about cooled ceiling is *not* provided here, thus, if required, some of the following references might rather be studied [2,6,8,20,27,48].

All preceding cooling strategies presented rely on an all-air system for cooling purposes, i.e., the loads are removed from the spaces by means on convective cooling. As most heat sources release the heat by convection and radiation, removing the load by taking advantage of both of these heat transfer mechanisms seems to be appropriate.

This way of cooling occupied spaces can be accomplished fairly easily by cooling the surfaces of a space with cooled water flowing through water pipes. For this purpose, the ceiling of a space is most suitable but, the walls or the floor can be used as well, although the cooling capacity of these is more limited. When the ceiling is used for cooling, the cooled water pipes can be put directly in the slab or can be attached to panels dropped from the ceiling. A cooled ceiling surface removes cooling loads both by convection and radiation. Depending on the construction, i.e., the kind of surface, the portion of radiation can be as high as 50 %. As this distinguishes a cooled ceiling system clearly from an all-air system, the term “radiant cooling system” is also been used. A water-cooled surface can only remove sensible loads, thus, fresh air needs to be supplied to provide both acceptable indoor air quality and remove the latent load, if necessary. Both, the way of removing the cooling load from the space and the reduced room air velocity when compared with a generic all-air system contribute to a higher level of thermal comfort. However, not only the improved thermal comfort is in favor for a radiant cooling system, but also the more advantageous operation costs.

Radiant cooling systems are usually operated with supply water temperatures of 15^{o}C or more. In any case, condensation at the cold parts of the construction has to be avoided. Dewpoint sensors shall be installed to either switch off the cooled water supply or increase the cooled water temperature when the dewpoint temperature of the room air approaches the temperature of the water supply. Common constructions of radiant cooled ceiling systems are able to remove sensible cooling loads of up to about 60-80 W/m^2. The better the construction of the cooled ceiling, the higher the supply water temperature might be, so that alternative strategies can be utilized more efficiently. Although, this issue has not been investigated in this study, it is a very important point, for radiant cooling applications. This topic is been discussed in the Chapter 14, “Final Discussion and Conclusions”.

Radiant cooled ceilings have been re-introduced in Europe more than 10 years ago and the usage for both new and retrofitted buildings is still increasing, however, there is almost no marked for cooled ceiling applications in the United States. To try to change this situation, an investigation comparing the energy efficiency of cooled ceiling system versus conventional all-air systems by means of computer simulation has been carried out recently by Stetiu [48].

The savings potential of a cooled ceiling system versus an all-air system is pointed out in the following sections using the methodology as well as some results of that study. Corina Stetiu was kind enough to conduct some further calculations regarding the European climates of Berlin and Locarno. Detailed information about the method and the simulation tools used can be found in [48].

13.1 Methodology

First, a simulation tool for calculating the performance of cooled ceiling, RADCOOL, was developed and validated with experimental results by Stetiu [45]. As the simulation of a space equipped with a cooled ceiling requires much more computation time than, e.g., a space with an all-air system, the comparison of the cooled ceiling system and the all-air VAV system presented is based on the energy characteristics of one particular room. The building described in Chapter 7 was also being used and the space "MBC2" (compare Chapter 7) has been chosen to represent the whole building best. The energy intensity to thermally condition this particular space and the peak power demand involved are used to compare the two different air-conditioning systems. The savings presented in the section "Results" are based on the energy use for the one space, however, the relative savings apply to the entire floor of the building in a comparable manner.

To limit the computation time with RADCOOL, the simulation was conducted for one *typical week* of the summer period. This typical week is a location-specific week that captures most of the variability in the energy consumption of the HVAC system over the cooling period. The estimate for the system energy use during this week can therefore be used to estimate the system's energy use during the entire cooling season. The typical week was chosen according to the average energy consumption for the entire cooling season and that for one week. For the climates of Berlin and Locarno, the typical week happens to be in May.

Since not only the energy consumption is an important issue to evaluate the performance of both the cooled ceiling system and the reference VAV system, but the electrical peak power demand as well, the period with the peak cooling load was additionally taken into consideration.

The comparison is based on an all-air system with variable volume flow, which corresponds to the reference VAV system, although the energy consumption and peak power demand are not exactly the same. The radiant cooling system under study is a panel type with a supply water temperature of 17.5^{o}C to 20^{o}C depending on the climatic conditions. A mechanical ventilation system provides the fresh air rate required for hygienical reasons.

13.2 Results

The results for the all-air system conditioning the space MBC2 were determined using DOE-2E, whereas, the energy and peak power demand of the radiant cooling system was calculated with RADCOOL. According to [48], no problems are involved using the results of these two different simulation tools. The performance of the all-air system was adjusted to match the indoor air conditions provided by the radiant cooling system. This fine-tuning process does not influence the characteristic of the all-air system simulated with DOE-2, although the results might be slightly different, when compared with the results of the reference VAV system presented in Chapter 8.4.2.

Figure 13.1 presents the system energy savings for 11 different climates within the United States (• from [48]) and for the two European climates of Berlin and Locarno. The savings in system energy use are shown as a function of the energy use of chillers and fans during the cooling period (May 1st through October 31st). The distribution of the results allow to draw a line (linear regression) to estimate the savings possible for other than the climates investigated. A range in which all the results can be found is shown as well. The energy consumption of the HVAC system can be reduced by about 12 % (Berlin) to 42 % (Scottsbluff and Phoenix) when a radiant cooling system is used instead of an all-air system. The major savings occur in the categories “fans” and “pumps”.

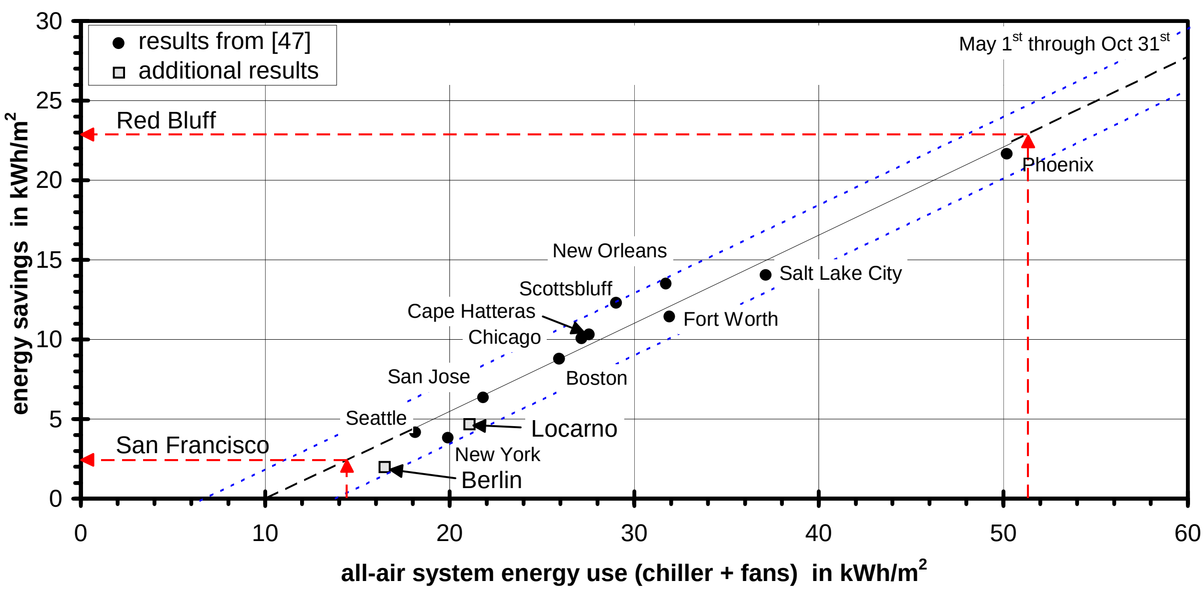

Figure 13.1 : Energy savings achievable with a radiant cooling system over an all-air system for different climates (cooling season) [48].

Figure 13.4 shows the system’s electrical peak power reduction possible if a radiant cooling system was utilized instead of an all-air system. With the climates investigated, the peak power demand can be lowered by 23 % (New York) to 37 % (Phoenix), mainly due to the reduced fan power demand.

Unfortunately, no simulation runs for the climates of Red Bluff and San Francisco could be made. However, knowing the energy use and peak power demand for the building in theses two regions, allows to estimate the savings in both categories. For this purpose, Figure 13.2 and Figure 13.2 can be used. The primary energy use of fans and chillers and the peak power demand of the system has to be determined and then, the savings can be estimated using the regression line in the mentioned figures. Table 13.1 summarizes the values required for this estimate, also presented in Figure 13.3 and Figure 13.2, as well as the results.

A radiant cooling system might offer savings in system energy consumption of about 17 % in a climate like San Francisco and 44 % in Red Bluff. The peak power demand of a radiant cooling system will also be lower than that of an all-air system. Peak power reductions of 28 % (San Francisco) to 36 % (Red Bluff) seem to be possible.

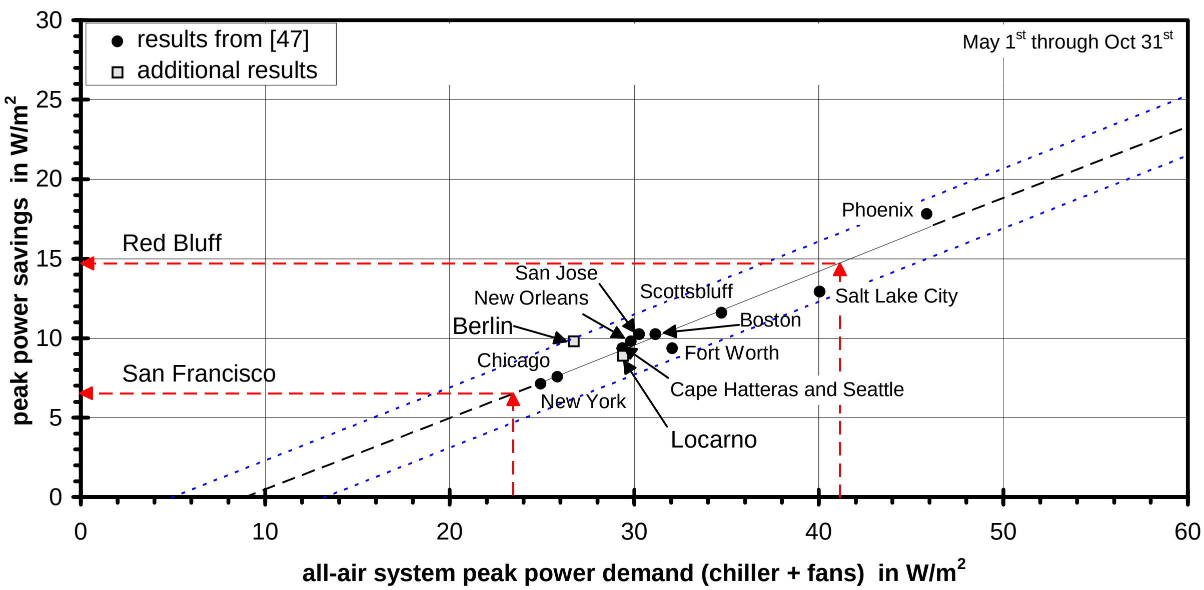

Figure 13.4 : Reduction in peak power demand with a radiant cooling system over an all-air system for different climates (cooling season) [48].

Table 13.1 : Energy intensity and peak power demand of the reference VAV system for the climates Red Bluff and San Francisco.

	chiller and fans					
	primary energy consumption			peak power demand		
climates	use	savings			savings	
	kWh/m^2	kWh/m^2	%	W/m^2	W/m^2	%
Red Bluff	51.4	23	44	41.2	14.8	36
San Francisco	14.3	2.5	17	23.6	6.5	28

13.3 Summary

The investigation of Stetiu [48] has shown that a radiant cooled ceiling combined with mechanical ventilation providing the required air exchange rate is able to improve the energy efficiency of an conventional all-air system in an modern office building. The system energy consumption of an all-air system can be reduced by about 12 % to 44 % and the electrical peak power demand can be decreased by 23 % to 37 %. The results gained by means of computer simulation allow to estimate the savings in system energy consumption and peak power demand if the characteristic energy data of an all-air system are known.

14. Final Discussion and Conclusions

The results presented in the preceding chapters show clearly that the energy efficiency of modern office building can be improved. In order to achieve most effective HVAC systems, certain conditions have to be fulfilled by the construction and usage of the building right from the beginning. The HVAC system has not to be blamed for mistakes made with the building design, however, this has and still will often been done.

First of all, the loads of a building have to be minimized by taking advantage of thermal storage capacity, shading devices, thermal insulation and energy efficient lighting and office equipment. Of course, these measures have been done, carefully considering the actual costs for the measures. However, not only the first and operation costs (energy use and maintenance) determine the energy efficiency of a building or HVAC system, respectively. The energy and costs involved to build the components of the system and the building as well as those to recycle or waste the materials play an important role. This issue will be investigated at the ETH in Zuerich.

The energy consumption of a modern office building can even be improved significantly by using a conventional HVAC system. If the design of the components is considering the actual demand (no over-sizing) and the performance is optimized, more favorable energy characteristics can be achieved when compared to today's standard. But still, conventional compressive cooling requires fairly high electrical power resulting in more expensive rates for electricity.

When compared with an optimized conventional HVAC system with compression chiller, using alternative cooling strategies instead can reduce both the primary energy consumption of a building and it's peak power demand. Depending on the climatic conditions, the most suitable cooling strategy can be chosen considering the results presented. Although, the primary energy use of a building or a system, respectively, should be decisive, more often the first and operation costs are. The impact on the environment, e.g., CO_2-generation, will become more important in the near future, so that alternative cooling strategies get more chance to be installed instead of conventional technology

The following sections deal with the annual energy costs for the cooling strategies investigated and finally, Chapter 14.2 compares both the annual energy costs and the primary energy consumption to make a final evaluation possible.

14.1 Annual operation costs

To compare the annual energy costs, two different assumptions regarding the energy prizes involved were made. The following rates (Table 14.1) are supposed to represent the current market for electricity and natural gas in California and Germany [29,55]. For Locarno, the German energy rates are considered, too, as no specific data for Switzerland were available. The rates listed in Table 14.1 include both the demand charges and the energy use costs, but do not include payback and interests, maintenance or other costs. It has to be noted that in this study, a reduction of the peak power demand did not result in more favorable rates for electricity. That means for practice, that even lower annual energy costs than shown might be accomplished with alternative cooling strategies reducing the electrical peak power demand.

Table 14.1 : Assumed prizes for energy in US$/kWh [29,55].

	California	Germany
electricity	0.08	0.15
natural gas	0.015	0.03
district heat	0.023	0.03

Using the rates shown in Table 14.1 in combination with the annual energy consumption for the different strategies and climates results in the annual operation costs presented in the figures of the following chapter.

14.2 Final comparison of annual energy costs and consumption

The following Figure 14.1 to 14.5 summarize the results of the different cooling strategies under study. The reference VAV system with a compressor-driven chiller (COP = 3.0) constitutes the base-case for this comparison, which means that, both the annual energy costs and the energy use of this system is set to 100 %. Unfortunately, the performance of the chiller used is not representing modern, highly developed compression chillers very well (compare footnote 13, page 30). Therefore, the energy consumption of the VAV system has been re-calculated using a compression chiller with a COP of 4.0. The results for all alternative cooling strategies being operated with an additional compression chiller, however, are still based on a COP of the compression chiller of 3.0.

First, for each of the climates, one figure is presenting the results of all cooling strategies investigated. This makes it possible to find out the most energy efficient strategy and also the one with the lowest energy costs for the particular climatic conditions. Additionally, the annual energy costs and the primary energy consumption of all strategies and climates are summarized in Table A21 (Appendix). Second, the peak power reduction possible with night ventilation, evaporative and desiccant cooling as well as using an absorption chiller instead of a compression chiller is presented in Figure 14.5. Then, short summaries for each of the cooling strategies under study are following (alphabetical order). Finally, some possible strategies, which might be worth to be investigated in the future, are shortly described.

Berlin and Kiel :

For the German climates only one figure (Figure 14.1) has been created, as the climatic conditions in the two regions do not differ significantly. Using other cooling strategies than compressive cooling, reduces the system energy costs and the energy consumption for heating, cooling and venting by only about 10 %. This is not surprising, as the portion of primary energy for cooling (compression chiller and cooling tower) only makes up about 11 to 14 % of the system primary energy use.

If waste heat with a suitable temperature level, e.g., > 70 °C, was available, both a single-stage absorption chiller and a desiccant cooling system could be used very advantageously. However, this energy source is rarely available, so that a combination of conventional cooling and mechanical night ventilation seems to be the most realistic alternative cooling strategy offering savings in primary energy consumption of 8 % and reducing the annual energy costs by about 9 %.

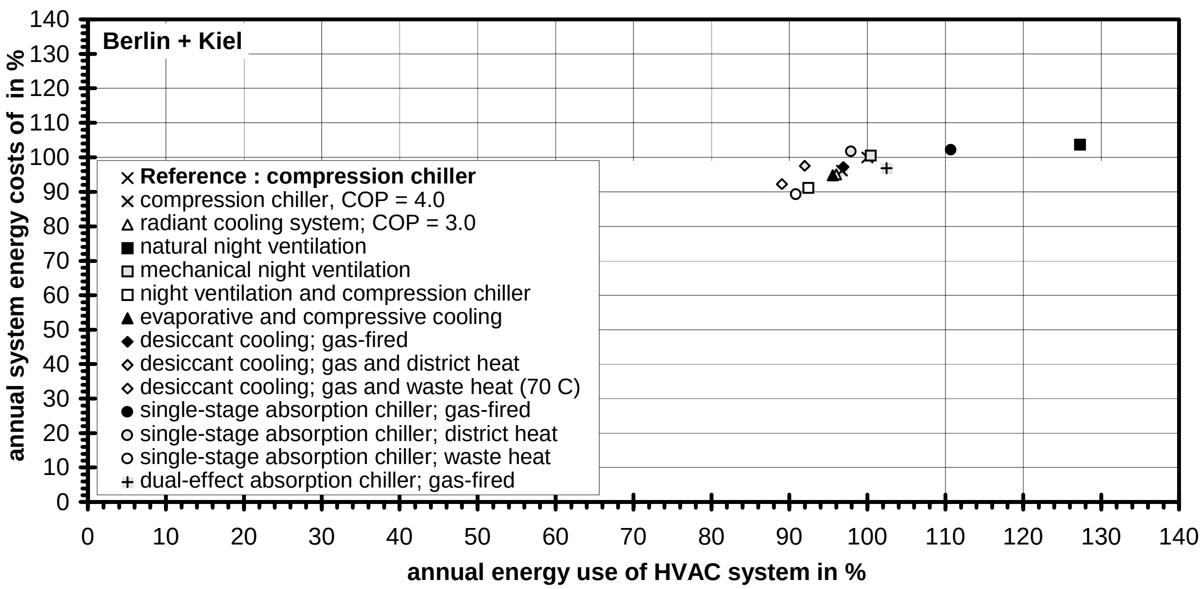

Figure 14.1 : Annual energy costs and primary energy consumption of the HVAC system with different cooling strategies in Berlin or Kiel.

Locarno :

The savings possible in a climate like Locarno are presented in Figure 14.2. Much higher savings in system energy can be accomplished with alternative cooling strategies than in the German climates, as about 33 % of the primary energy consumed by the reference VAV system is used for cooling. Again, waste heat utilized for either desiccant cooling or a single-stage absorption chiller provides the most energy efficient system performance. Savings of about 30 % both in costs and energy use can be achieved with waste heat usage. Considering only strategies not using waste heat, desiccant cooling appears to be the most economic solution in this case. If district heat was available at a temperature of 70 ^{o}C, the primary energy consumption of the system could be reduced by 25 % when compared with the reference VAV system. If only gas and electricity could be used to operate the HVAC system, a combination of evaporative pre-cooling and compressive cooling or gas-fired desiccant cooling constitute the most energy efficient way to thermally condition the building investigated. These two different set-ups provide primary energy savings of about 10 % considering the climatic conditions of Locarno.

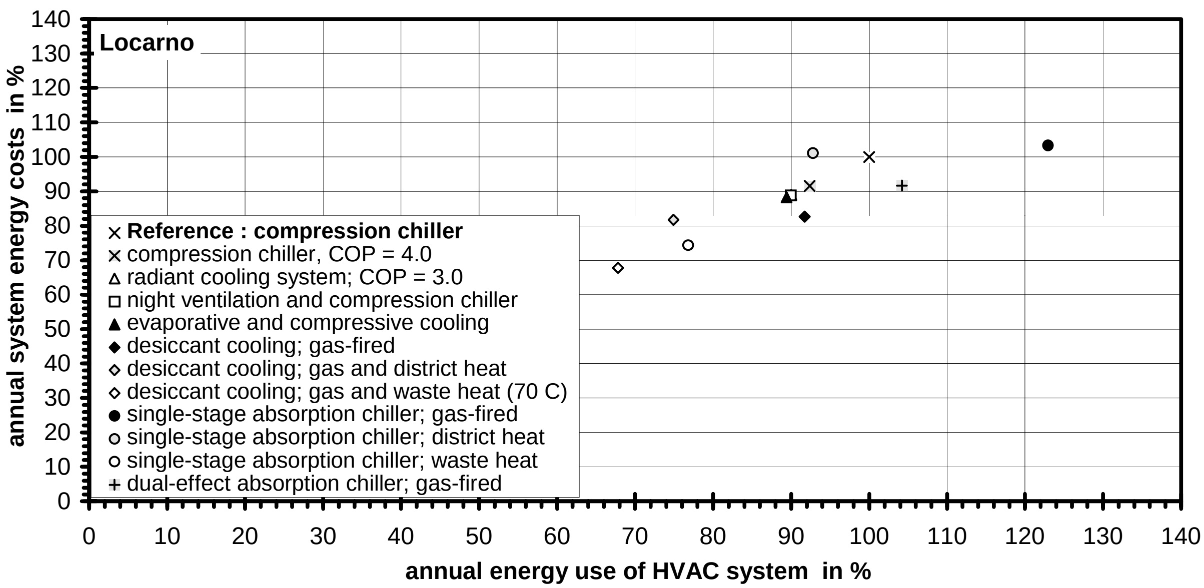

Figure 14.2 : Annual energy costs and primary energy consumption of the HVAC system with different cooling strategies in Locarno.

Red Bluff :

With the climatic conditions of Red Bluff, about 52 % of the primary energy use of the HVAC system are consumed for cooling purposes. Here, evaporative cooling combined with a conventional cooling coil (compressive cooling) offers by far the highest savings both in primary energy consumption and energy costs (Figure 14.3). Even higher financial benefits are possible as no price reduction for electricity is considered, although the significantly lower peak power demand with evaporative and compressive cooling would certainly lead to more favorable electricity rates. However, the energy costs and use of the system can be halved with this strategy when compared with the reference VAV system.

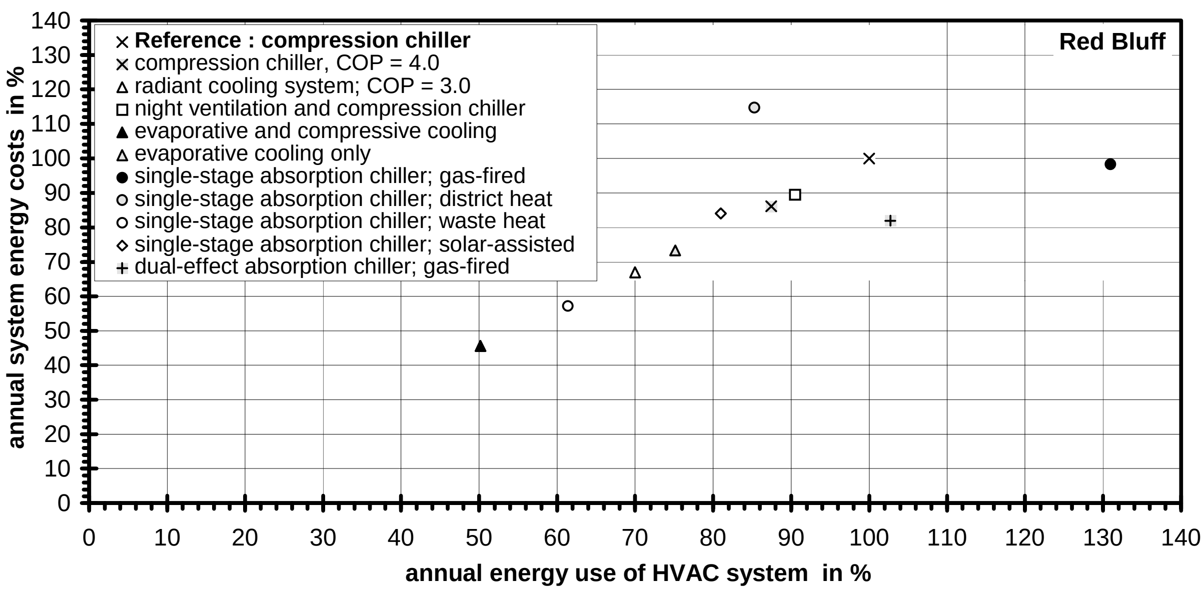

Figure 14.3 : Annual energy costs and primary energy consumption of the HVAC system with different cooling strategies in Red Bluff.

San Francisco :

The climate of San Francisco seems to be designated for taking advantage of the temperate outside air[45], especially at night. As no remarkable heating energy use is involved here, the savings in cooling and fan energy provided with natural night ventilation are much higher than the slightly increased energy for heating purposes (see Chapter 9.1). If natural night ventilation could be accomplished as assumed in this study, the energy cost could be cut by half and the primary energy use could be reduced by almost 30 % when compared with the reference VAV system. Second best alternative is evaporative cooling offering savings in primary energy and annual energy costs of about 25 % each.

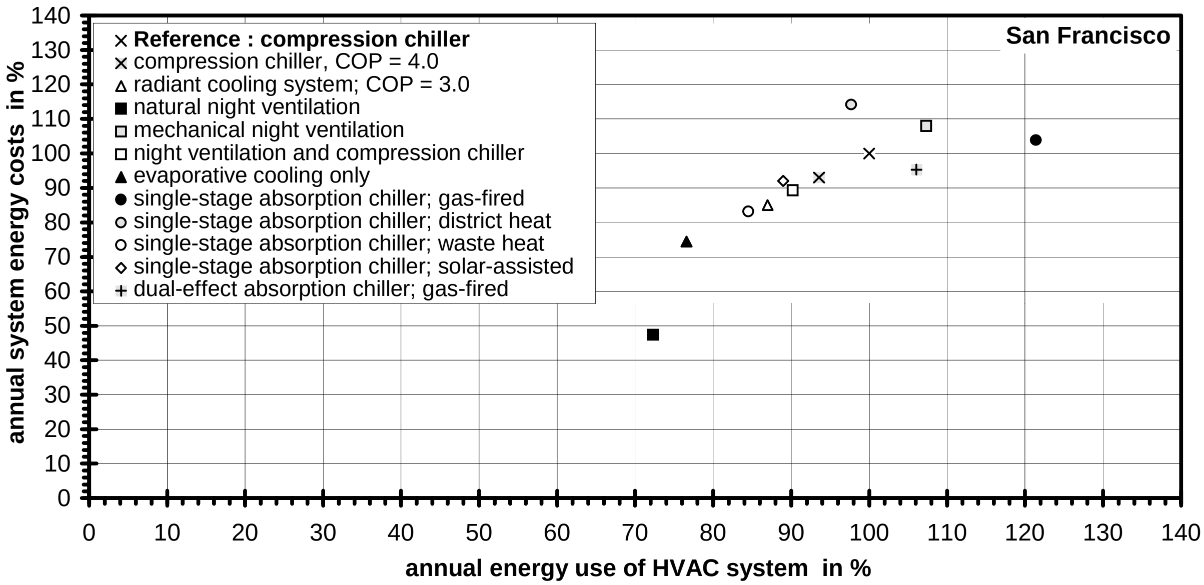

Figure 14.4: Annual energy costs and primary energy consumption of the HVAC system with different cooling strategies in San Francisco.

Figure 14.5 summarizes the peak power savings with some of the alternatives under study. It has to be noted that two variations of night ventilation are presented for the climate of San Francisco and also two variations of evaporative cooling for the climate of Red Bluff. The peak power demand of the reference VAV system can easily be reduced utilizing an alternative cooling strategy instead.

[45] The temperate air temperature throughout the year in San Francisco and the East Bay Area, respectively, is colloquially named *natural air-conditioning system*.

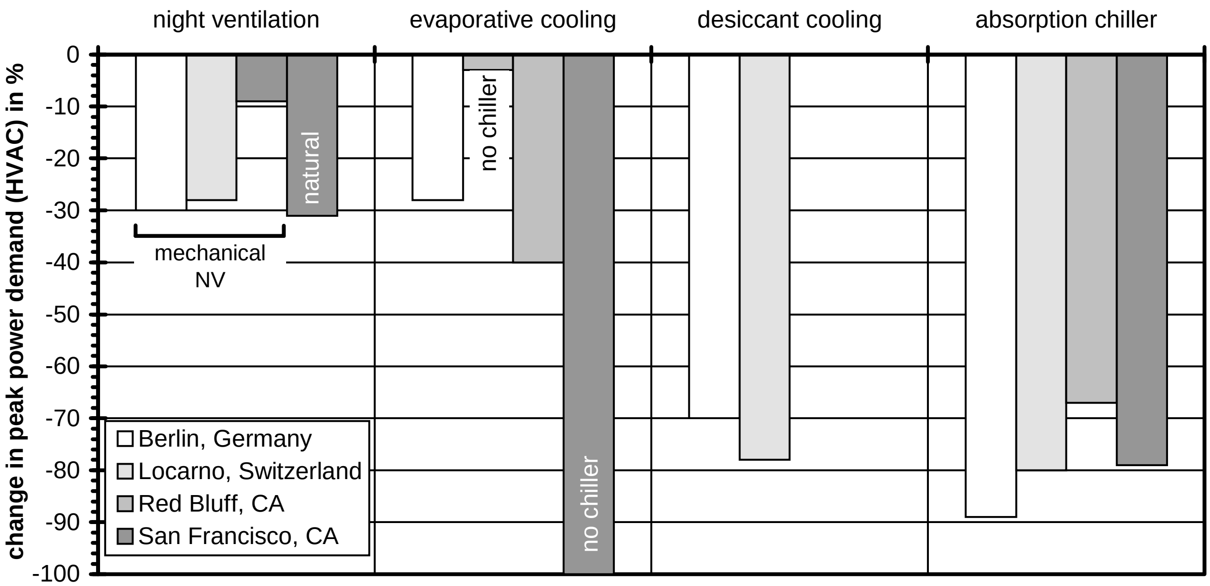

Figure 14.5 : Reduction in peak power demand for some of the alternative strategies investigated when compared with the reference VAV system.

<u>Absorption chiller :</u>

Absorption chillers are the only alternative able to completely substitute a conventional compression chiller and to provide the same chilled water temperatures. Especially, when the supply air has to be dehumidified by means of cooling below the dewpoint temperature, a mechanical chiller is required. Absorption chillers do not use refrigerants harmful to the environment (atmosphere) and use much less electrical energy than the conventional counterpart.

However, for all but one of the climates under study, operating a gas-fired single-stage absorption chiller is more expensive than a conventional compression chiller. In Red Bluff, the annual energy costs for the system can be reduced by about 2 % when a single-stage gas-fired absorption chiller replaces the compression chiller. The primary energy consumption of a gas-fired single-stage absorption chiller is disadvantageous anyhow. This changes if waste heat, district heat or heat from solar-collectors could be used to operate the absorption chiller.

Although, the primary energy use of a gas-fired dual-effect absorption chiller is still less favorable for all climatic conditions investigated, using these chillers, offers savings in annual energy costs of 3 % (Berlin) to 18 % (Red Bluff) of the energy expenses for conventional compressive cooling (COP =3.0). Assuming a better compression chiller (COP = 4.0), no benefits of a dual-effect chiller are possible anymore.

Considering these results with absorption chillers, favorable usage seems only to be possible in conjunction with waste or district heat or with additional solar-collectors. Gas-fired absorption chillers, either utilizing a single-stage or a dual-effect process, do not offer any primary energy savings over an modern compression chiller.

Compression chiller :

Conventional compression chillers are without any doubt are very reliable, highly developed and last but not least, the common choice to provide chilled water with no technical problems except, the high electricity use and peak power demand as well as the problems related to the refrigerants used.

Using high efficient compression chillers (COP = 4.0), which are not over-sized, can provide a much lower primary energy consumption than common today. However, the building's design and the operation of the system has to be carried out properly. Accepting floating indoor air temperatures and maybe even some hours of insufficient cooling capacity, e.g., when the windows have been opened individually, although the outside air temperature is very high, can save a fair amount of primary energy.

As it is usually more difficult to replace a technology proven to be mature, skillfully making use of highly developed compressor-driven chiller running on non harmful refrigerants, e.g., NH_3, propane or butane, might be a convinient and energy efficient solution for HVAC applications. However, the relatively high electricity consumption and the high peak power demand do not favor this conventional cooling process.

Desiccant cooling :

In regions with significant humidity ratios, dehumidification might be requested, mainly to provide a high level of thermal comfort. If radiant cooled ceilings are used to remove the sensible cooling load, dehumidification can be required to obtain an appropriate cooling performance of the ceiling.

Desiccant cooling seems to be the only alternative cooling strategy providing a competitive dehumidification capability. The components utilized with desiccant cooling do not represent exotic devices, in contrary, the desiccant regenerator, the heat exchanger and the humidifier are highly developed HVAC components, which are used for air-handling purposes for years. Only the combination of desiccant regenerator and indirect and direct evaporative coolers is not (yet) that common. With this strategy no materials harmful to the environment are involved.

The performance of the desiccant cooling system investigated proves that energy efficient air-conditioning with dehumidification can be accomplished without operating a classic refrigeration process. With a gas-fired system, only small primary energy savings over a conventional system are possible, but fossil fuel, preferably natural gas, or waste or district heat mostly replace the electrical energy used by the latter. Heat from solar-collectors might be suitable as well, but such a system was not studied yet. Due to fuel switching, the peak power demand of a desiccant cooling system is much lower than that of a conventional cooling system with compression chiller. The primary energy consumption of a desiccant cooling system can be improved significantly (compared to a gas-fired set-up), if waste or district heat were available. Optimization of the performance of the desiccant wheel, the control strategy and the kind of heat provided will probably result in more energy efficient desiccant cooling systems.

Evaporative Cooling :

If the climatic conditions do not require dehumidification, evaporative cooling can be used to thermally condition a building. Although, the cooling performance of a stand-alone system can be sufficient even in very hot climates, the increased fan power might compensate for the electrical cooling energy saved. Therefore, the required supply airflow has to be considered carefully when designing the duct work to make savings both in primary energy and peak power demand possible.

Combining evaporative coolers with a conventional cooling coil, is a smart method to reduce primary energy use and the peak power demand. This strategy is eligible both for climates with dehumidification needs and those without. However, combining evaporative coolers and conventional compressive cooling still involves problems with refrigerants.

Night Ventilation :

This strategy is very qualified to reduce the primary energy consumption of HVAC system, both conventional and alternative ones. A remarkable thermal storage capacity is even more important for this cooling method than probably for the others as the cooling only can be obtained by means of pre-cooling the building structure at night and charging it with heat released during occupancy.

If natural night ventilation is considered, the uncertainty of the air actually exchanged only by means of natural pressure and temperature differences has to be taken into account. The stack effect can be used to increase night-venting a building, but the cooling capacity might change from day to day. Additionally, the heating energy consumption will increase (compared to a conventional HVAC system) as no heat from the return air can be recovered. Mechanically supplied night air can solve these problems, however, the fan energy consumption might compensate for the cooling provided by the outside air.

Although, night ventilation was only investigated in conjunction with compressive cooling, taking advantage of the colder outside air at night will very likely result in energy savings when other cooling strategies, e.g., evaporative cooling, are used. In many cases, the cooling potential of the outside might not be sufficient to cool a building sufficiently, however, it certainly contributes to lower the primary energy consumption and the peak power demand of an additionally operated HVAC system.

Radiant cooled ceiling :

The energy consumption and peak power demand of an all-air HVAC system can be improved when radiant cooled ceilings are used to remove the sensible cooling load. A very welcome "side-effect", actually this is very often the reason to put such a system in modern office buildings, is the improvement of thermal comfort. Draft and uncomfortable temperature are less likely to occur in spaces with radiant cooled ceilings than, e.g., in spaces with an all-air system.

Cooled ceilings can be combined with many cooling sources, even alternative methods not providing suitable chilled water temperatures for dehumidification can be used to supply cold water for the ceiling cycle.

14.3 Future Research

This section is supposed to point out other ways to save energy or cooling strategies to replace or at least support compressor-driven chillers in HVAC systems. Most likely, the investigation of the following methods would be performed best by means of computer simulation.

Cooled ceiling with direct usage of cooling tower :

A cooling tower ought to be able to sufficiently provide cooled water to a radiant cooled ceiling. This set-up has only a chance of success when the supply water temperature required for the cooled ceiling is significantly higher than the wet-bulb temperature plus the approach of the cooling tower. However, a cooling tower in combination with a radiant cooling system is only able to reduce the cooling energy for sensible cooling. The latent load has still to be removed, if required, with another method.

Desiccant cooling and radiant cooling with cooling tower :

For removing the latent load from a space which sensible load is mainly removed by a cooled ceiling receiving cooled water from a cooling tower, desiccant cooling might be useful. As the supply airflow can be reduced to the fresh air requirement, a fairly small desiccant cooling unit is needed. However, the performance of the desiccant remains the same offering small saving potentials in primary energy when gas-fired desiccant cooling is compared with compressive cooling. Again, making use of waste heat or district heat improves the energy efficiency of such a system.

Desiccant cooling with intercooler (outside air) :

The most apparent problem with desiccant cooling are the relatively high regeneration temperatures required to sufficiently dehumidify the supply air. Lower regeneration temperatures result in a better performance of the desiccant wheel and make usage of waste heat at low temperatures possible.

As the desiccant requires a certain regeneration temperature to retain the moisture from the supply air, only a two-stage desiccant cooling system seem to be able to run the system on lower temperatures. The supply air leaving the first desiccant wheel could be inter-cooled by outside air (high efficient heat exchanger) before entering the second desiccant wheel. This set-up should result in lower regeneration temperatures reducing the gas consumption if a gas-fired heater is operated. If waste heat and gas were used in series, the additional gas heating would be required at fewer hours of the cooling season.

Electrochromatic windows :

Electrochromatic windows offer the opportunity to reduce the cooling loads due to solar radiation without having shading devices, like exterior blinds. The transmission of the electrochromtic coating of the windowpane can be controlled by applying a low voltage to the coating. This changes the visible transmittance as well as the shading coefficient influencing the solar gains. Optimizing the control of electrochromatic windows contributes to reduce the solar gains without having mechanical problems with handling and the maintenance of the shading devices and allow a comfortable possibility for day-lighting. The performance of different types of electrochromatic windows and the capability to reduce both the peak power demand and the energy consumption of a building with such windows are currently investigated at LBL [44,49].

Ground coupling :

In winter, the temperature of the soil is usually higher than the outside air and in summer, the ground temperature is usually lower than the outside air temperature. This circumstance can be used to pre-heat the outside air in winter or pre-cool the supply air in summer. A heat exchanger embedded in the soil underneath a building can be utilized.

A simulation tool to calculate the temperature of an air volume flow drawn through such an earth-register was developed at the ETH Zuerich and simulation runs are currently carried out at LBL. Results with this strategy will hopefully be available in Fall 1997.

Phase change material :

A high thermal storage capacity is the key to reduce the cooling peak load. Especially, American buildings are usually built without a remarkable building mass. Putting a material, which changes its phase (solid-liquid) at a certain temperature within the comfort range, into building materials, like sheet rock, increases the thermal storage capacity of the walls without increasing the mass of the building or the construction. Paraffin can be used for such applications, however, only prototypes of wallboard have been manufactured in the United States so far.

Investigations of the performance of phase change material have been conducted for residential buildings at LBL and the results are encouraging [62,63]. An application for office buildings, e.g., for partition walls, might lead to additional savings in energy use and peak power demand. There are no results available yet.

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17. Appendix

17.1 Climatic parameters

Table A1 : Annual statistics on the dry-bulb temperature for the five climates under study (based on the corresponding TRY or TMY)

dry-bulb temperature	location									
	Berlin		Kiel		Locarno		Red Bluff		San Francisco	
in ^oC	h	Σ %	h	Σ %	h	Σ %	h	Σ %	h	Σ %
-10	0	0	7	0.1	2	0	0	0	0	0
-8	11	0.1	22	0.3	8	0.1	0	0	0	0
-6	23	0.4	49	0.9	52	0.7	0	0	0	0
-4	69	1.2	61	1.6	89	1.7	4	0	0	0
-2	177	3.2	245	4.4	263	4.7	27	0.4	0	0
0	236	5.9	292	7.7	347	8.7	71	1.2	0	0
2	720	14.1	928	18.3	681	16.5	198	3.4	18	0.2
4	992	25.4	1046	30.3	728	24.8	396	7.9	82	1.1
6	853	35.2	826	39.7	505	30.5	448	13.1	264	4.2
8	950	46.0	797	48.8	692	38.4	770	21.8	551	10.4
10	654	53.5	522	54.7	567	44.9	592	28.6	757	19.1
12	867	63.4	821	64.1	715	53.1	749	37.2	1982	41.7
14	817	72.7	844	73.7	723	61.3	694	45.1	2157	66.3
16	610	79.7	761	82.4	584	68.0	509	50.9	1094	78.8
18	749	88.2	876	92.4	774	76.8	688	58.7	813	88.1
20	370	92.4	331	96.2	540	83.0	502	64.5	420	92.9
22	331	96.2	189	98.4	548	89.2	608	71.4	332	96.7
24	163	98.1	88	99.4	428	94.1	557	77.8	161	98.5
26	78	99.0	32	99.7	246	96.9	336	81.6	48	99.1
28	70	99.8	14	99.9	201	99.2	394	86.1	53	99.7
30	18	100	9	100	56	99.9	251	89.0	15	99.9
32	2	100	0	100	9	100	280	92.2	8	99.9
34	0	100	0	100	2	100	277	95.3	5	100
36	0	100	0	100	0	100	155	97.1	0	100
38	0	100	0	100	0	100	116	98.4	0	100
40	0	100	0	100	0	100	65	99.2	0	100
42	0	100	0	100	0	100	41	99.6	0	100
44	0	100	0	100	0	100	16	99.8	0	100
46	0	100	0	100	0	100	8	99.9	0	100
48	0	100	0	100	0	100	8	100	0	100
$\bar{t}_{dry-bulb}$	**9.5 ^oC**		**8.7 ^oC**		**11.1 ^oC**		**16.5 ^oC**		**13.0 ^oC**	

Table A2 : Annual statistics on the humidity ratio for the five climates under study (based on the corresponding TRY or TMY)

humidity ratio	location									
	Berlin		Kiel		Locarno		Red Bluff		San Francisco	
in g/kg	h	Σ %	h	Σ %	h	Σ %	h	Σ %	h	Σ %
0	0	0	0	0	0	0	0	0	0	0
2	111	1.3	132	1.5	293	3.3	140	1.6	19	0.2
3	524	7.2	485	7.0	1030	15.1	564	8.0	51	0.8
4	1247	21.5	1390	22.9	1321	30.2	1143	21.1	256	3.7
5	1484	38.4	1600	41.2	1094	42.7	1470	37.9	765	12.5
6	1289	53.1	1176	54.6	746	51.2	1747	57.8	1478	29.3
7	1019	64.8	861	64.4	713	59.3	1386	73.6	1800	49.9
8	912	75.2	921	74.9	746	67.8	986	84.9	2046	73.2
9	741	83.6	794	84.0	604	74.7	583	91.5	1860	94.5
10	669	91.3	608	90.9	629	81.9	395	96.1	401	99.0
11	508	97.1	440	96.0	589	88.6	189	98.2	84	100
12	192	99.3	248	98.8	519	94.6	93	99.3	0	100
13	52	99.9	78	99.7	276	97.7	45	99.8	0	100
14	12	100	27	100	105	98.9	17	100	0	100
15	0	100	0	100	71	99.7	2	100	0	100
16	0	100	0	100	15	99.9	0	100	0	100
17	0	100	0	100	8	100	0	100	0	100
18	0	100	0	100	1	100	0	100	0	100
$\overline{x}$	**6.1 g/kg**		**6.1 g/kg**		**6.4 g/kg**		**5.8 g/kg**		**6.8 g/kg**	

Table A3 : Annual statistics on the global solar radiation for the five climates under study (based on the corresponding TRY or TMY)

global solar radiation	**location**									
	Berlin		**Kiel**		**Locarno**		**Red Bluff**		**San Francisco**	
in W/m²	h	Σ %	h	Σ %	h	Σ %	h	Σ %	h	Σ %
0	4091	46.7	44148	47.4	4215	48.1	4276	48.8	4239	48.4
100	1829	67.6	2001	70.2	1601	66.4	920	59.3	938	59.1
200	1045	79.5	996	81.6	731	74.7	617	66.4	591	65.8
300	606	86.4	476	87.0	597	81.6	489	71.9	483	71.4
400	381	90.8	290	90.3	439	86.6	398	76.5	405	76.0
500	268	93.8	220	92.8	349	90.5	418	81.3	437	81.0
600	201	96.1	185	94.9	283	93.8	326	85.0	380	85.3
700	192	98.3	173	96.9	235	96.5	327	88.7	343	89.2
800	118	99.7	147	98.6	204	98.8	294	92.1	312	92.8
900	28	100	92	99.6	93	99.9	329	95.8	326	96.5
1000	1	100	31	100	13	100	304	99.3	306	100
1100	0	100	1	100	0	100	62	100	0	100
$\bar{I}_{a,global}$	**210 W/m²**		**213 W/m²**		**260 W/m²**		**406 W/m²**		**400 W/m²**	

Table A4 : Annual statistics on the wet-bulb temperature for the five climates under study (based on the corresponding TRY or TMY)

wet-bulb temperature	**location**									
	Berlin		**Kiel**		**Locarno**		**Red Bluff**		**San Francisco**	
in ^{o}C	h	Σ %	h	Σ %	h	Σ %	h	Σ %	h	Σ %
-10	0	0	5	0.1	2	0	0	0	0	0
-8	14	0.2	42	0.5	18	0.2	0	0	0	0
-6	44	0.7	64	1.3	61	0.9	0	0	0	0
-4	123	2.1	95	2.4	132	2.4	5	0.1	0	0
-2	218	4.6	301	5.8	415	7.2	45	0.6	0	0
0	358	8.6	407	10.4	579	13.8	139	2.2	11	0.1
2	928	19.2	1070	22.6	839	23.4	386	6.6	60	0.8
4	1198	32.9	1172	36.0	863	33.6	753	15.2	220	3.3
6	871	42.9	824	45.4	536	39.7	764	23.9	439	8.3
8	964	53.9	809	54.7	754	48.3	1093	36.4	1212	22.2
10	688	61.7	564	61.1	609	55.2	923	46.9	1559	40.0
12	1039	73.6	1035	72.9	781	64.1	1138	59.9	2382	67.2
14	960	84.5	1105	85.5	926	74.7	1161	73.1	1998	90.0
16	688	92.4	725	93.8	724	83.0	788	82.1	670	97.6
18	542	98.6	409	98.5	871	92.9	809	91.4	198	99.9
20	102	99.7	110	99.7	432	97.9	443	96.4	11	100
22	23	100	23	100	160	99.7	274	99.6	0	100
24	0	100	0	100	25	100	39	100	0	100
26	0	100	0	100	3	100	0	100	0	100
$\bar{t}_{wet-bulb}$	**7.4 ^{o}C**		**7.1 ^{o}C**		**8.2 ^{o}C**		**10.3 ^{o}C**		**10.3 ^{o}C**	

Table A5 : Annual statistics on the dewpoint temperature for the five climates under study (based on the corresponding TRY or TMY)

dewpoint temperature	**location**									
	Berlin		**Kiel**		**Locarno**		**Red Bluff**		**San Francisco**	
in ^oC	h	Σ %	h	Σ %	h	Σ %	h	Σ %	h	Σ %
-10	47	0.5	64	0.7	145	1.7	61	0.7	0	0
-8	60	1.2	60	1.4	147	3.3	78	1.6	10	0.1
-6	105	2.4	106	2.6	327	7.1	138	3.2	14	0.3
-4	224	5.0	172	4.6	437	12.1	271	6.3	12	0.4
-2	421	9.8	455	9.8	610	19.0	496	11.9	75	1.3
0	699	17.8	677	17.5	728	27.3	498	17.6	119	2.6
2	1018	29.4	1066	29.7	726	35.6	824	27.0	267	5.7
4	1055	41.4	1161	42.9	769	44.4	1168	40.3	614	12.7
6	913	51.8	858	52.7	546	50.6	1303	55.2	1053	24.7
8	944	62.6	733	61.1	658	58.1	1334	70.4	1625	43.3
10	901	72.9	879	71.1	736	66.5	1063	82.6	1891	64.8
12	891	83.1	973	82.2	697	74.5	746	91.1	2288	91.0
14	819	92.4	753	90.8	781	83.4	463	96.4	705	99.0
16	541	98.6	562	97.2	782	92.3	216	98.8	87	100
18	112	99.9	211	99.7	500	98.0	83	99.8	0	100
20	10	100	30	100	146	99.7	18	100	0	100
22	0	100	0	100	23	100	0	100	0	100
24	0	100	0	100	2	100	0	100	0	100
$\bar{t}_{dewpoint}$	**5.6 ^oC**		**5.7 ^oC**		**5.5 ^oC**		**4.9 ^oC**		**8.0 ^oC**	

17.2 Dry-bulb and coincident wet-bulb temperatures

Table A6 : TRY 3 (Berlin), annual hours between 7^{oo} and 19^{oo} h sorted by dry-bulb and wet-bulb temperatures ($t_{dry,max}$ = 30.0 ^oC)

	wet-bulb temperatures in ^oC																	
dry	**26**	**25**	**24**	**23**	**22**	**21**	**20**	**19**	**18**	**17**	**16**	**15**	**14**	**13**	**12**	**11**	**10**	**sum**
30	0	0	0	0	0	0	6	0	0	0	0	0	0	0	0	0	0	**6**
29	0	0	0	0	0	2	7	5	0	0	0	0	0	0	0	0	0	**14**
28	0	0	0	0	3	0	2	16	3	2	0	0	0	0	0	0	0	**26**
27	0	0	0	0	1	2	4	16	10	10	0	0	0	0	0	0	0	**43**
26	0	0	0	0	0	0	1	5	8	6	3	0	0	0	0	0	0	**23**
25	0	0	0	0	0	0	6	8	17	10	1	3	0	0	0	0	0	**45**
24	0	0	0	0	0	0	2	4	16	20	6	3	1	0	0	0	0	**52**
23	0	0	0	0	0	0	0	8	25	23	4	11	6	0	0	0	0	**77**
22	0	0	0	0	0	0	1	1	16	46	14	19	16	3	0	0	0	**116**
21	0	0	0	0	0	0	0	1	11	24	7	11	9	6	4	1	0	**75**
20	0	0	0	0	0	0	0	0	8	41	22	34	21	12	12	3	1	**156**
19	0	0	0	0	0	0	0	0	7	31	26	32	26	30	10	2	1	**166**
18	0	0	0	0	0	0	0	0	3	17	25	47	35	41	30	1	2	**200**
17	0	0	0	0	0	0	0	0	0	13	16	56	44	37	42	6	5	**220**
16	0	0	0	0	0	0	0	0	0	0	2	24	19	22	20	8	5	**103**
15	0	0	0	0	0	0	0	0	0	0	0	13	43	66	42	18	18	**200**
14	0	0	0	0	0	0	0	0	0	0	0	0	19	65	55	21	28	**181**
13	0	0	0	0	0	0	0	0	0	0	0	0	0	17	73	24	49	**138**
12	0	0	0	0	0	0	0	0	0	0	0	0	0	0	37	41	54	**119**
11	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	4	44	**8**
10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	33	**0**
sum :	**0**	**0**	**0**	**0**	**4**	**4**	**29**	**64**	**124**	**243**	**126**	**253**	**239**	**299**	**325**	**129**	**240**	**1968**

Table A7 : TRY 1 (Kiel) : annual hours between 7^{oo} and 19^{oo} h sorted by dry-bulb and wet-bulb temperatures ($t_{dry,max}$ = 29 ^oC)

	wet-bulb temperature in ^oC																
dry	**26**	**25**	**24**	**23**	**22**	**21**	**20**	**19**	**18**	**17**	**16**	**15**	**14**	**13**	**12**	**11**	**10**	**sum**
29	0	0	0	0	0	1	4	4	0	0	0	0	0	0	0	0	0	**9**
28	0	0	0	0	0	2	2	0	0	0	0	0	0	0	0	0	0	**4**
27	0	0	0	0	0	0	7	2	0	0	0	0	0	0	0	0	0	**9**
26	0	0	0	0	0	0	3	2	0	0	0	0	0	0	0	0	0	**5**
25	0	0	0	0	0	1	3	6	8	5	0	1	2	0	0	0	0	**26**
24	0	0	0	0	0	1	8	7	6	12	3	3	2	0	0	0	0	**42**
23	0	0	0	0	0	0	5	8	6	6	2	2	7	3	0	0	0	**39**
22	0	0	0	0	0	0	5	22	15	8	2	9	5	2	0	0	0	**68**
21	0	0	0	0	0	0	2	13	9	8	3	5	4	0	0	0	0	**44**
20	0	0	0	0	0	0	0	7	26	24	15	28	14	3	2	5	0	**129**
19	0	0	0	0	0	0	0	0	10	48	30	42	35	12	5	4	1	**190**
18	0	0	0	0	0	0	0	0	3	45	50	81	67	18	7	2	3	**275**
17	0	0	0	0	0	0	0	0	0	10	27	81	54	61	10	4	4	**251**
16	0	0	0	0	0	0	0	0	0	0	0	33	38	46	20	1	6	**139**
15	0	0	0	0	0	0	0	0	0	0	0	21	53	102	52	14	7	**256**
14	0	0	0	0	0	0	0	0	0	0	0	0	20	73	75	22	19	**212**
13	0	0	0	0	0	0	0	0	0	0	0	0	0	20	77	40	43	**177**
12	0	0	0	0	0	0	0	0	0	0	0	0	0	0	35	40	75	**115**
11	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	28	**6**
10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	10	**0**
sum:	**0**	**0**	**0**	**0**	**0**	**5**	**39**	**71**	**83**	**166**	**132**	**306**	**301**	**340**	**283**	**135**	**196**	**1996**

Table A8 : Locarno : annual hours between 7^{oo} and 19^{oo} h sorted by dry-bulb and wet-bulb temperatures ($t_{dry,max}$ = 34 ^{o}C)

	wet-bulb temperature in ^{o}C																
dry	**26**	**25**	**24**	**23**	**22**	**21**	**20**	**19**	**18**	**17**	**16**	**15**	**14**	**13**	**12**	**11**	**10**	**sum**
34	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	**1**
33	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	**1**
32	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	**1**
31	0	0	0	2	2	1	2	0	0	1	0	0	0	0	0	0	0	**8**
30	0	0	0	5	1	1	4	0	1	2	0	0	0	0	0	0	0	**14**
29	0	0	3	7	7	5	13	6	2	2	1	1	1	1	0	0	0	**49**
28	0	0	0	6	4	9	36	20	1	6	1	0	2	0	0	0	0	**85**
27	0	0	0	2	8	8	29	37	20	5	2	2	2	0	0	0	0	**115**
26	0	0	0	0	2	4	14	46	42	11	2	10	6	2	0	0	0	**139**
25	0	0	0	1	1	3	14	46	41	23	2	13	8	3	0	0	0	**155**
24	0	0	0	0	1	4	6	35	48	37	10	16	8	4	1	0	0	**170**
23	0	0	0	0	1	2	5	28	20	44	20	29	14	4	9	2	0	**180**
22	0	0	0	0	0	0	5	22	19	29	11	35	21	7	16	7	1	**179**
21	0	0	0	0	0	0	2	16	16	36	15	35	32	18	10	8	0	**196**
20	0	0	0	0	0	0	0	10	22	31	7	38	31	37	9	9	3	**203**
19	0	0	0	0	0	0	0	2	34	20	15	35	28	24	22	6	6	**192**
18	0	0	0	0	0	0	0	0	8	46	15	26	22	32	23	13	10	**198**
17	0	0	0	0	0	0	0	0	0	7	25	35	21	18	19	6	12	**137**
16	0	0	0	0	0	0	0	0	0	0	2	41	26	26	27	6	24	**134**
15	0	0	0	0	0	0	0	0	0	0	0	17	30	30	29	7	22	**120**
14	0	0	0	0	0	0	0	0	0	0	0	0	24	41	15	16	15	**112**
13	0	0	0	0	0	0	0	0	0	0	0	0	0	15	30	15	19	**75**
12	0	0	0	0	0	0	0	0	0	0	0	0	0	0	12	29	39	**70**
11	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	54	**4**
10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	26	**0**
sum	**1**	**2**	**3**	**23**	**27**	**37**	**130**	**268**	**274**	**300**	**128**	**333**	**276**	**262**	**222**	**126**	**231**	**2538**

Table A9 : Red Bluff : annual hours between 7^{oo} and 19^{oo} h sorted by dry-bulb and wet-bulb temperatures ($t_{dry,max}$ = 48 °C)

	wet-bulb temperature in °C																	
dry	**26**	**25**	**24**	**23**	**22**	**21**	**20**	**19**	**18**	**17**	**16**	**15**	**14**	**13**	**12**	**11**	**10**	**sum**
48	0	0	0	2	2	0	0	0	0	0	0	0	0	0	0	0	0	**4**
47	0	0	0	1	3	0	0	0	0	0	0	0	0	0	0	0	0	**4**
46	0	0	0	1	1	0	0	0	0	0	0	0	0	0	0	0	0	**2**
45	0	0	0	1	1	0	0	4	0	0	0	0	0	0	0	0	0	**6**
44	0	0	0	2	1	1	3	0	0	0	0	0	0	0	0	0	0	**7**
43	0	0	0	0	1	1	7	0	0	0	0	0	0	0	0	0	0	**9**
42	0	0	0	2	5	2	6	5	0	0	0	0	0	0	0	0	0	**20**
41	0	0	0	0	3	0	1	4	0	0	0	0	0	0	0	0	0	**8**
40	0	0	0	4	8	0	6	7	1	1	0	0	0	0	0	0	0	**27**
39	0	0	0	1	12	3	11	10	8	3	0	0	0	0	0	0	0	**48**
38	0	0	0	6	4	5	13	8	8	0	0	0	0	0	0	0	0	**44**
37	0	0	0	6	8	7	14	14	14	2	1	0	0	0	0	0	0	**66**
36	0	0	0	1	5	3	11	7	6	4	1	0	0	0	0	0	0	**38**
35	0	0	1	4	8	10	16	28	19	15	2	6	0	0	0	0	0	**109**
34	0	0	0	0	5	6	20	24	27	18	2	7	1	0	0	0	0	**110**
33	0	0	0	0	3	4	15	25	44	32	5	7	2	1	0	0	0	**138**
32	0	0	0	0	1	1	21	20	27	28	7	5	6	3	0	0	0	**119**
31	0	0	0	0	3	4	12	18	28	26	12	4	8	2	0	0	0	**117**
30	0	0	0	0	3	2	11	17	31	21	14	13	9	2	0	0	0	**123**
29	0	0	0	0	2	4	3	16	22	23	23	20	6	5	0	0	0	**124**
28	0	0	0	0	0	0	3	12	12	25	22	23	26	1	1	0	0	**125**
27	0	0	0	0	0	1	0	13	19	22	13	35	25	15	2	0	0	**145**
26	0	0	0	0	0	0	1	2	6	11	13	34	28	19	5	1	0	**121**
25	0	0	0	0	0	0	1	1	14	12	18	38	32	17	6	1	0	**141**
24	0	0	0	0	0	0	0	0	3	10	17	26	26	32	13	2	5	**131**
23	0	0	0	0	0	0	0	1	0	10	6	37	35	29	22	8	6	**156**
22	0	0	0	0	0	0	0	0	1	6	5	20	29	45	19	11	10	**147**
21	0	0	0	0	0	0	0	0	2	4	5	17	37	40	32	7	14	**151**
20	0	0	0	0	0	0	0	0	2	6	5	14	33	39	31	16	13	**162**
19	0	0	0	0	0	0	0	0	0	1	0	13	28	35	35	5	25	**122**
18	0	0	0	0	0	0	0	0	0	2	4	4	22	32	24	23	24	**134**
17	0	0	0	0	0	0	0	0	0	0	0	1	19	16	38	19	32	**112**
16	0	0	0	0	0	0	0	0	0	0	0	4	10	15	39	17	27	**102**
15	0	0	0	0	0	0	0	0	0	0	0	0	10	13	34	15	22	**87**
14	0	0	0	0	0	0	0	0	0	0	0	0	1	6	24	14	35	**59**
13	0	0	0	0	0	0	0	0	0	0	0	0	0	1	28	6	42	**41**
12	0	0	0	0	0	0	0	0	0	0	0	0	0	0	11	9	35	**29**
11	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	27	**0**
10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	8	**0**
sum	**0**	**0**	**1**	**31**	**79**	**54**	**175**	**236**	**294**	**282**	**175**	**328**	**393**	**368**	**364**	**154**	**325**	**3088**

Table A10 : San Francisco : annual hours between 7^{oo} and 19^{oo} h sorted by dry-bulb and wet-bulb temperatures ($t_{dry,max}$ = 34 °C)

	wet-bulb temperature in °C																	
dry	**26**	**25**	**24**	**23**	**22**	**21**	**20**	**19**	**18**	**17**	**16**	**15**	**14**	**13**	**12**	**11**	**10**	**sum**
34	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	**1**
33	0	0	0	0	0	0	0	0	0	2	2	0	0	0	0	0	0	**4**
32	0	0	0	0	0	0	0	0	0	1	1	0	0	0	0	0	0	**2**
31	0	0	0	0	0	0	0	1	2	2	1	0	0	0	0	0	0	**6**
30	0	0	0	0	0	0	1	2	3	2	1	2	0	0	0	0	0	**11**
29	0	0	0	0	0	0	1	3	3	0	0	1	0	0	0	0	0	**8**
28	0	0	0	0	0	0	0	2	3	8	1	5	2	0	0	0	0	**21**
27	0	0	0	0	0	0	0	1	8	6	4	4	4	3	0	0	0	**30**
26	0	0	0	0	0	0	0	1	3	12	6	1	1	2	0	0	0	**26**
25	0	0	0	0	0	0	0	1	5	12	9	1	2	0	1	0	0	**31**
24	0	0	0	0	0	0	0	0	12	25	10	10	4	2	1	1	0	**66**
23	0	0	0	0	0	0	0	0	9	27	23	20	5	3	2	2	0	**93**
22	0	0	0	0	0	0	0	0	8	36	37	44	7	5	3	1	1	**142**
21	0	0	0	0	0	0	0	0	0	13	28	99	18	6	6	2	1	**174**
20	0	0	0	0	0	0	0	0	0	12	17	126	35	9	3	1	2	**204**
19	0	0	0	0	0	0	0	0	0	0	10	144	97	26	9	4	8	**294**
18	0	0	0	0	0	0	0	0	0	1	6	55	183	85	19	7	4	**363**
17	0	0	0	0	0	0	0	0	0	0	2	19	165	112	31	21	11	**371**
16	0	0	0	0	0	0	0	0	0	0	0	19	108	166	113	39	46	**484**
15	0	0	0	0	0	0	0	0	0	0	0	6	76	153	126	56	71	**473**
14	0	0	0	0	0	0	0	0	0	0	0	0	15	135	156	74	93	**454**
13	0	0	0	0	0	0	0	0	0	0	0	0	0	16	111	91	123	**309**
12	0	0	0	0	0	0	0	0	0	0	0	0	0	0	18	42	180	**102**
11	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	47	**0**
10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	15	**0**
sum	**0**	**0**	**0**	**0**	**0**	**0**	**2**	**11**	**56**	**159**	**159**	**556**	**722**	**723**	**599**	**341**	**602**	**3669**

17.3 Energy Statistics

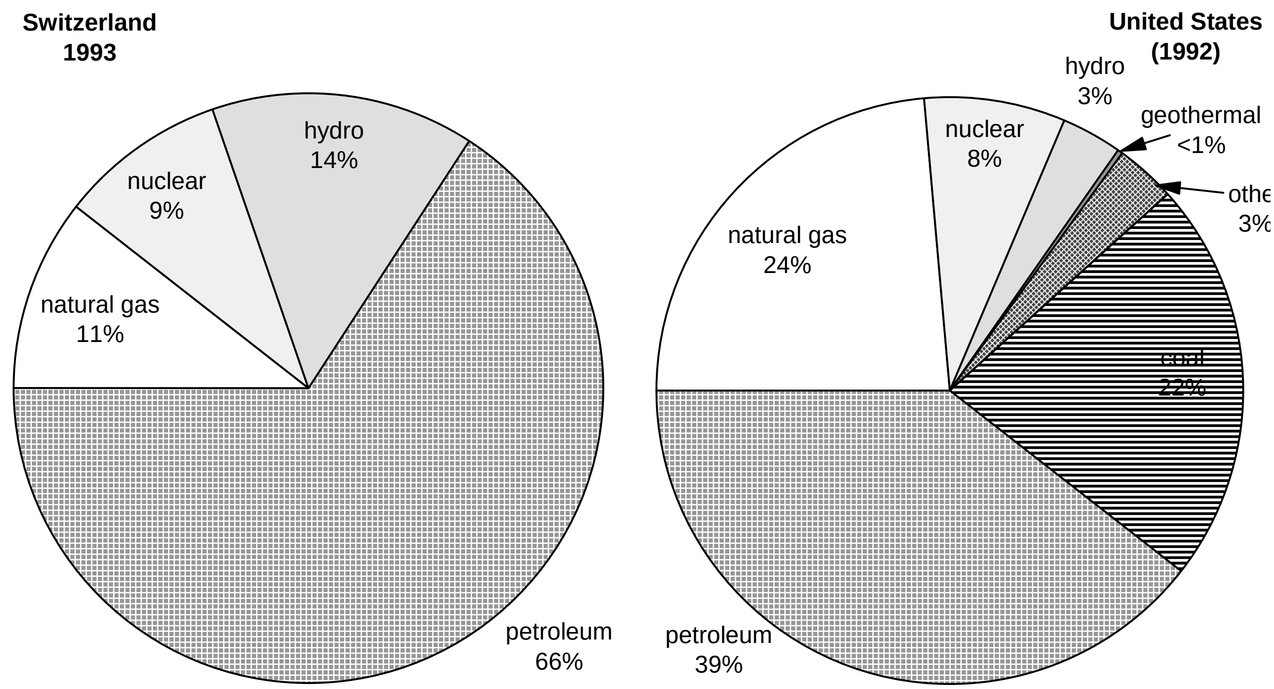

Figure A1 : Total energy demand of Switzerland and the United States in % by energy source [55,57].

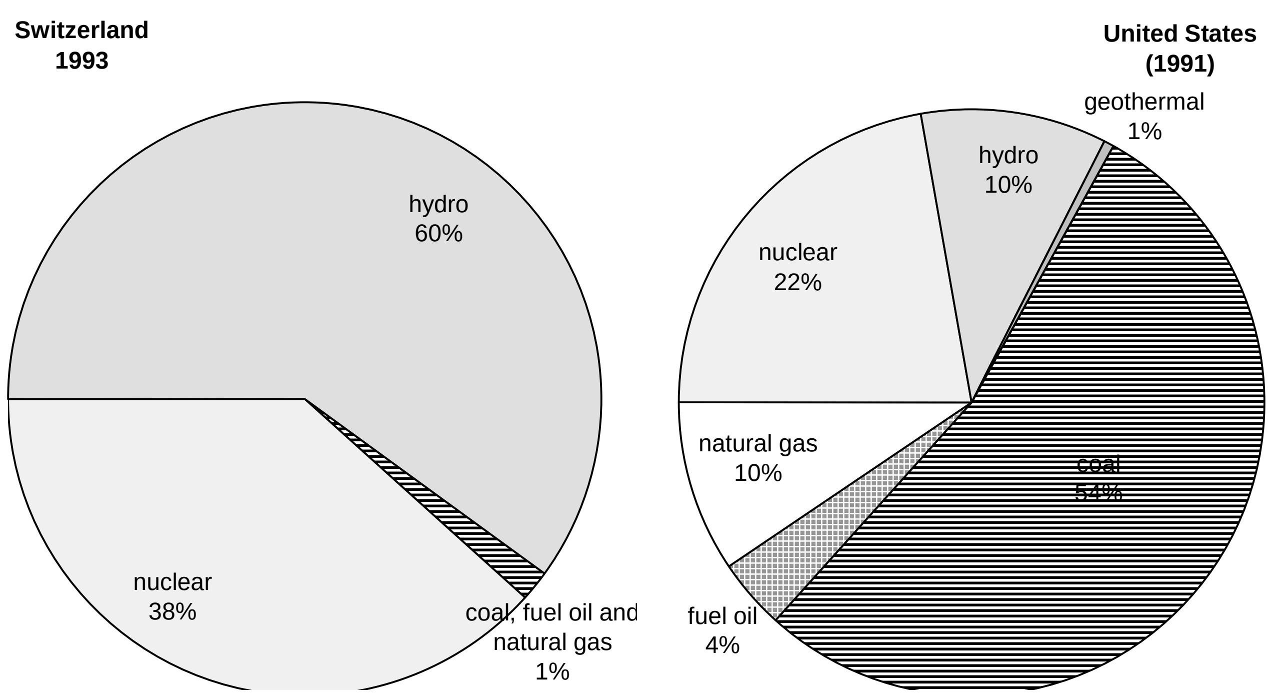

Figure A2 : Electricity generation in % for Switzerland and the United States by primary energy source [55,57].

17.4 Building and loads

17.4.1 Data of the building components

17.4.1.1 Opaque components

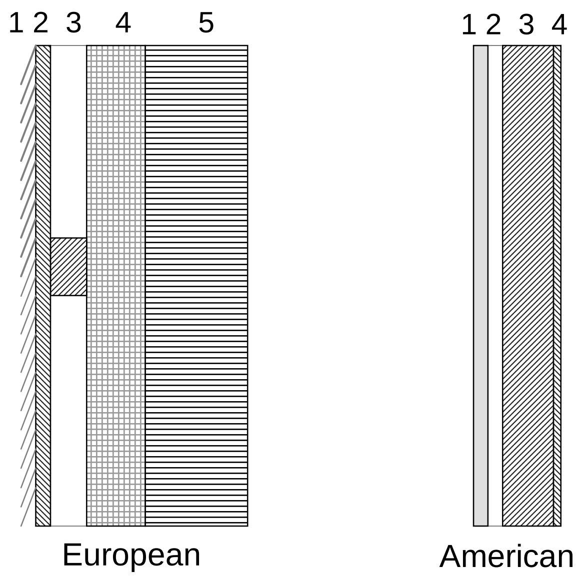

Figure A3 : Structure of the European and American exterior wall.

Table A11 : Data of the European exterior wall.

		thickness [m]	conductivity [W/m K]	density [kg/m^3]
1	**siding**	0.02	0.2	700
2	**sheathing**	0.05	0.2	600
3	**wooden laths and air space**	0.08	0.45*)	1.2
4	**insulation layer**	0.12	0.036	60
5	**concrete**	0.20	1.8	2400

*) the shares of wood and air space are considered

Table A12 : Data of the American exterior wall-element

		thickness [m]	conductivity [W/m K]	density [kg/m^3]
1	**glazing**	0.006	0.8	2464
2	**air space**	0.05	0.025 [1)]	1.2
3	**fiberboard**	0.102	0.056	48
4	**sheet rock**	0.016	0.16	800

1) An additional convective heat transfer has to be considered, thus, a combined heat resistance is been used insted : $R_\lambda = 0.177\ m^2\ K/W$

Table A13 : Data of the floor/ceiling

	thickness [m]	conductivity [W/m K]	density [kg/m^3]
concrete	0.12	1.8	2400
absorption layer	0.04	0.036	60
concrete	0.20	1.8	2400

Table A14 : Data of the interior walls

	thickness [m]	conductivity [W/m K]	density [kg/m^3]
sheet rock	0.01	0.21	900
air space	0.06	0.034	60
sheet rock	0.01	0.21	900

17.4.1.2 The window element

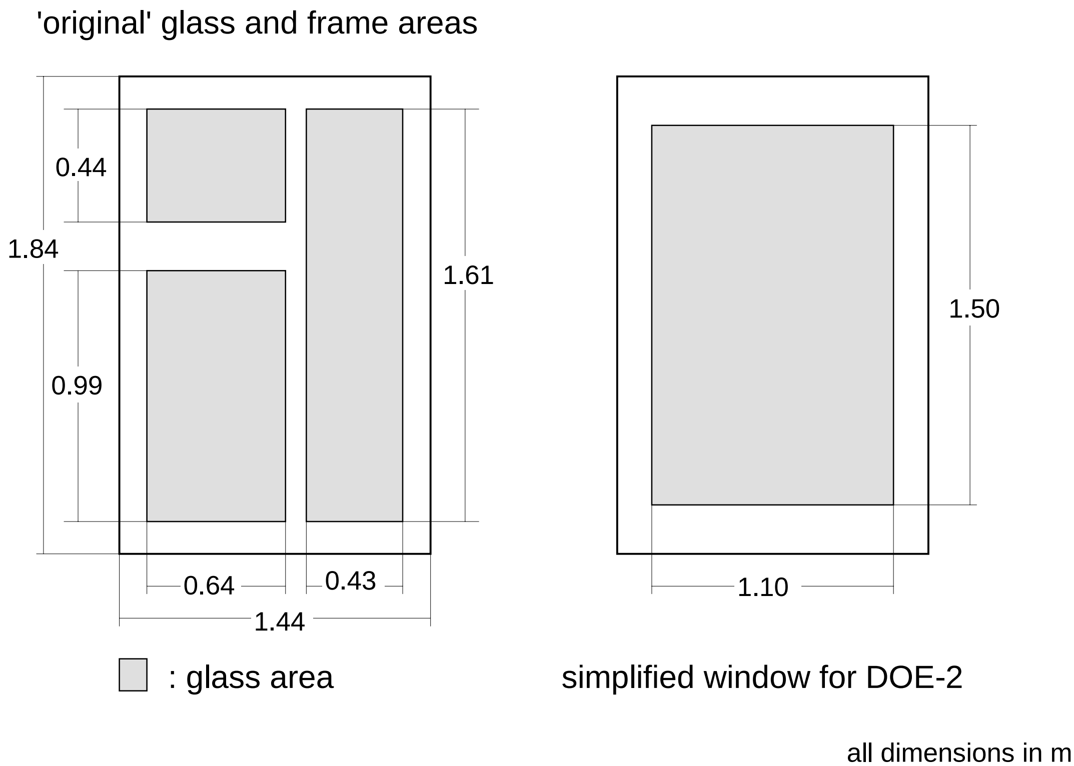

Figure A4 : Glass and frame areas of the window element.

17.4.2 Interior thermal loads

Table A15 : Interior loads

	number of occupants *) **)	max. heat from lighting [W]	heat from equipment [W]
single-person office	1	144	275
two-persons office	2	144	275
four-persons office	4	288	650
conference room	10	324	130
cafeteria	20	280	0
hallway	0	1000	0

*) the entire heat release (sensible and latent) of one person was taken as 120 W [15].

) the particular room is not entirely occupied during the whole time (compare **Error! Reference source not found.). The shown number of persons presents an occupancy of 100%.

17.5 Details on the HVAC systems investigated

17.5.1 Reference VAV system

Table A16 : Pressure drops for the HVAC components of the reference VAV system

	component	Δp_{max} for max. airflow in Pa
supply air side	prefilter	60
	heat recovery	75
	preheater	35
	humidifier	75
	cooling coil	50
	heating coil	35
	filter	85
	duct [1)]	300
sum for DOE-2 inputfile :		**715**
return air side	prefilter	60
	heat recovery	75
	duct	250
sum for DOE-2 inputfile :		**385**

1) includes the VAV box and air diffuser

17.5.2 Evaporative cooling system

Table A17 : Pressure drops for the HVAC components of the evaporative cooling system

	component	Δp_{max} for max. airflow in Pa
supply air side	prefilter	60
	heat exchanger [1)]	100
	preheater	35
	direct evap. cooler [2)]	75
	cooling coil	50
	heating coil	35
	filter	85
	duct [3)]	300
sum for DOE-2 inputfile :		**740**
return air side	prefilter	60
	direct evap. cooler [2)]	75
	heat exchanger	100
	duct	250
sum for DOE-2 inputfile :		**485**

1) is used for heat recovery in wintertime
2) is used as humidifier in wintertime
3) includes the VAV-box and the diffuser
4) This pressure drop for the supply air duct applies only to the variations with direct and direct evaporative coolers and additional cooling coil. The respective pressure drop is 665 Pa for Locarno (no direct evaporative cooler) and 690 Pa for San Francisco (no cooling coil.

17.5.3 Desiccant Cooling

Table A18 : Pressure drops for the HVAC components of the desiccant cooling system

	component	Δp_{max} for max. airflow in Pa
supply air side	prefilter	60
	adsorption wheel [1)]	200 [2)]
	heat exchanger [1)]	100
	preheater	35
	direct evap. cooler [3)]	75
	cooling coil	50
	heating coil	35
	filter	85
	duct [4)]	300
sum for DOE-2 inputfile :		**940**
return air side	prefilter	60
	direct evap. cooler	75
	heat exchanger [1)]	100
	gas-fired heater	35
	adsorption wheel [1)]	200 [2)]
	duct	250
sum for DOE-2 inputfile :		**720**

1) is used for heat recovery in wintertime
2) manufacturer's data
3) is used as humidifier in wintertime
4) includes the VAV-box and the diffuser

17.6 Performance data for the desiccant wheel

17.6.1 Regeneration temperature

The regeneration temperature, t_{regen}, is set according to the following equations (return air is used for regeneration and the return airflow almost equals the supply airflow) :

for $x_{enter} \geq 10$ g/kg :

$$t_{regen} = 1.5 * (t_{enter} - 34) + 61.367 - 4054 * x_{enter} + 402887 * x_{enter}^2 \quad \text{(A1)}$$

for $x_{enter} < 10$ g/kg :

$$t_{regen} = 60.0 + 1.5 * (t_{enter} - 34) \quad \text{(A2)}$$

with : t_{enter} : entering dry-bulb temperature in ^oC
x_{enter} : entering humidity ratio (outside air) in kg/kg

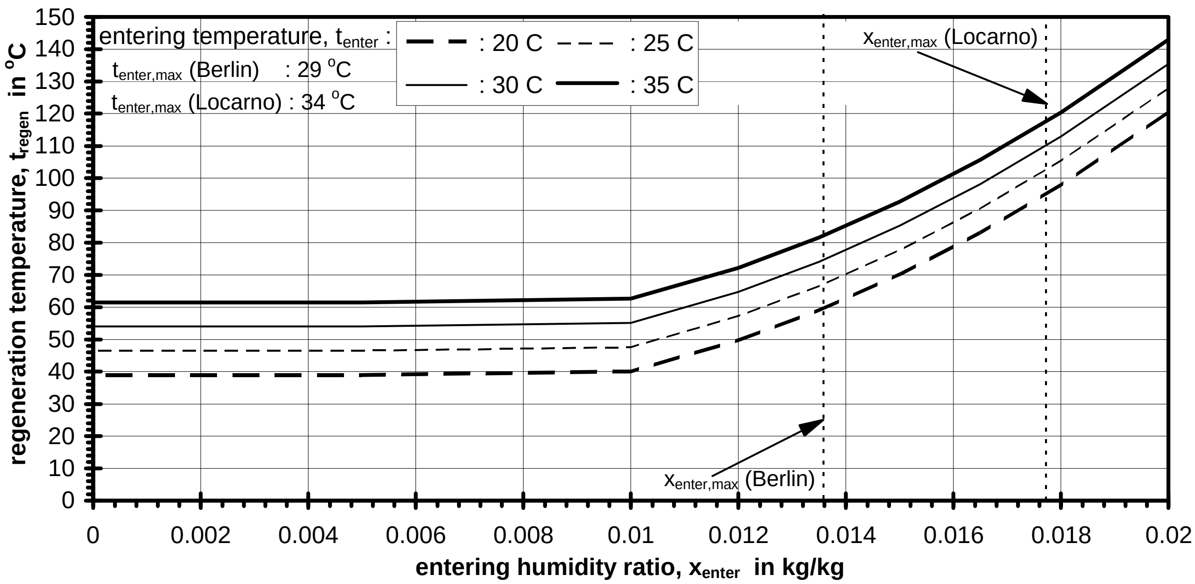

Figure A5 : Regeneration temperature used in the desiccant model as a function of the outside air humidity ratio, x_{enter}, and the entering dry-bulb temperature.

17.6.2 Temperature leaving the desiccant wheel

Based on the performance data for 75 and 95 ^{o}C the temperature of the airflow leaving the desiccant wheel is determined as a function of the outside air humidity ratio, x_{enter}, and dry-bulb temperature, t_{enter}, and the regeneration temperature, t_{regen} (see Chapter 17.6.1):

$$t_{exit} = 0.99753 * (t_{regen}^{0.5}) * (x_{enter}^{0.5}) + 0.37019 * (t_{regen}^{0.25})/(x_{enter}^{0.25}) * t_{enter} \qquad (A3)$$

with : t_{enter} : entering dry-bulb temperature in ^{o}C
x_{enter} : entering humidity ratio (outside air) in g/kg
t_{regen} : regeneration temperature in ^{o}C

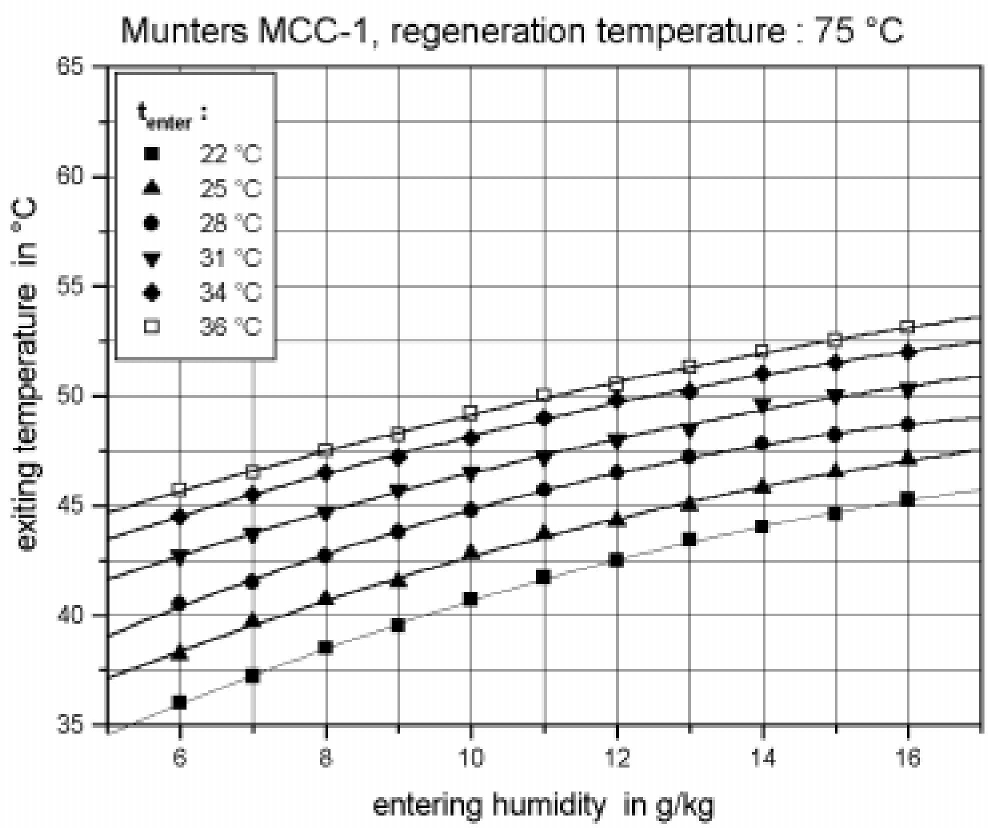

Figure A6 : Temperature leaving the desiccant wheel as a function of the entering humidity ratio and with a regeneration temperature of 75 ^{o}C (manufacturer's data)

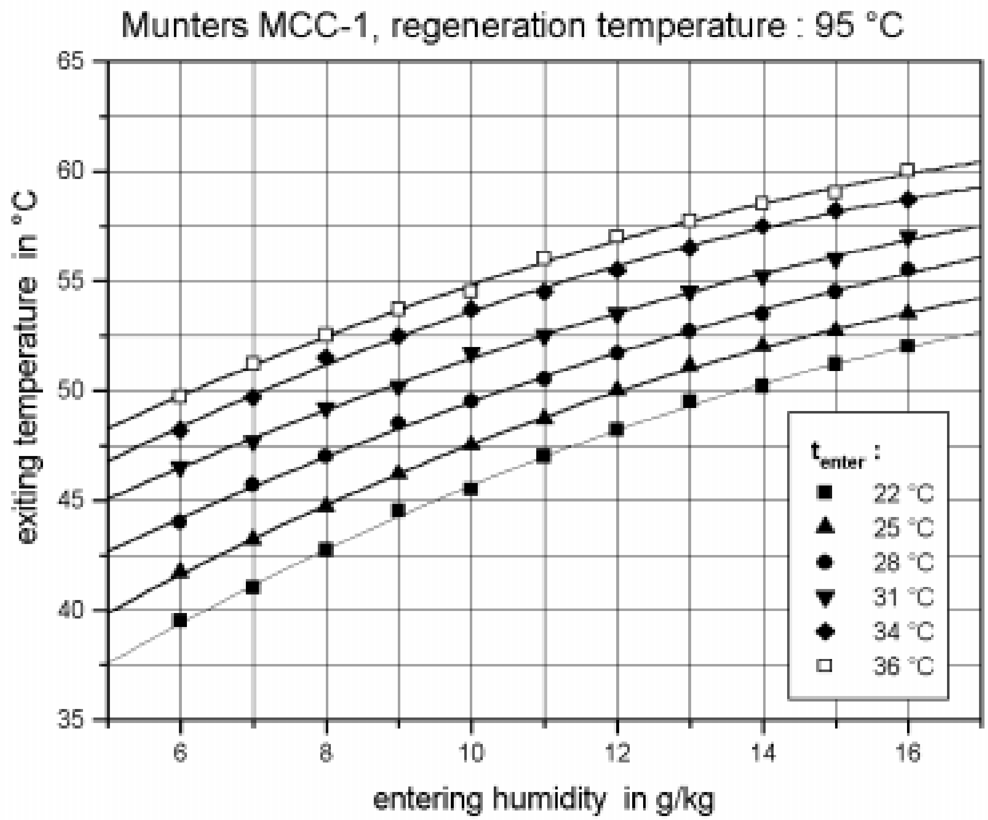

Figure A7 : Temperature leaving the desiccant wheel as a function of the entering humidity ratio and with a regeneration temperature of 95 ^{o}C (manufacturer's data)

17.6.3 Humidity ratio leaving the desiccant wheel

The humidity ratio of the air leaving the desiccant wheel, x_{exit}, is determined by inter- and extrapolation based on the performance of the wheel (moisture removal) with 75 and 95 ^oC regeneration temperature.

for t_{regen} = 75 ^oC :

$$x_{exit} = 0.0012 * t_{enter}^2 + 0.0086\ t_{enter} * x_{enter} + 0.0177 * x_{enter}^2 \quad \text{in g/kg} \qquad \text{(A4)}$$

for t_{regen} = 95 ^oC :

$$x_{exit} = 0.0027 * t_{enter}^2 + 0.01\ t_{enter} * x_{enter} + 0.0123 * x_{enter}^2 \quad \text{in g/kg} \qquad \text{(A5)}$$

with : t_{enter} : entering dry-bulb temperature in ^oC
x_{enter} : entering humidity ratio (outside air) in g/kg
t_{regen} : regeneration temperature in ^oC

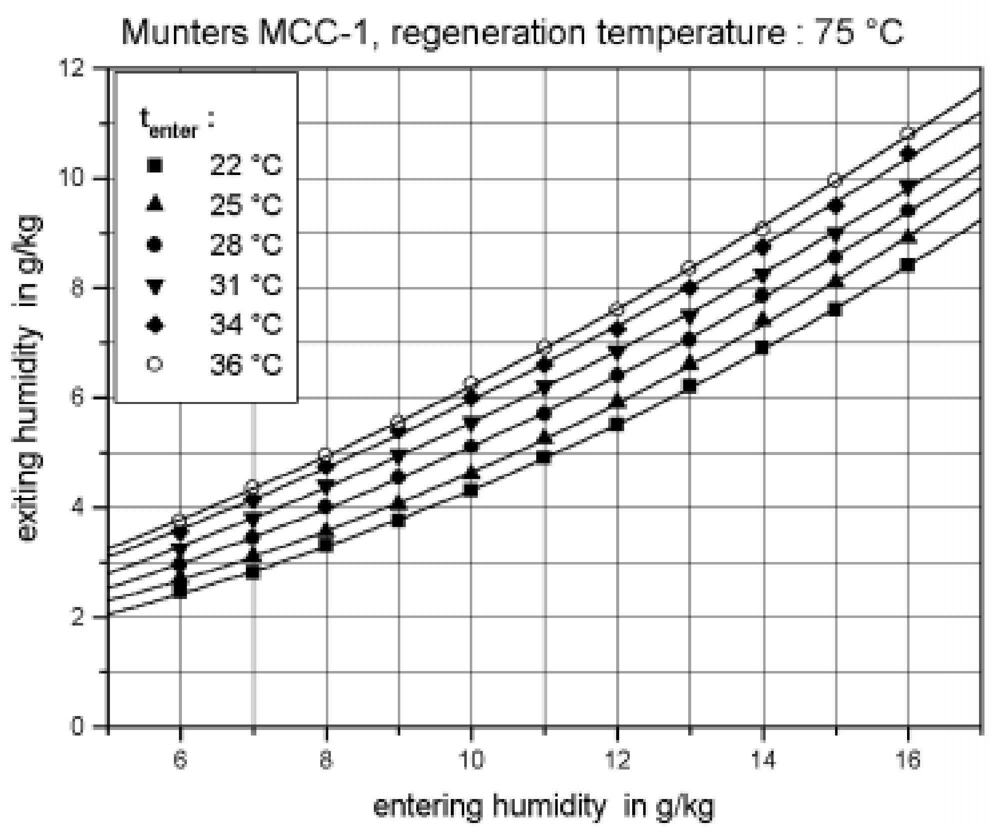

Figure A8 : Humidity ratio leaving the desiccant wheel as a function of the entering humidity ratio and with a regeneration temperature of 75 ^oC (manufacturer's data)

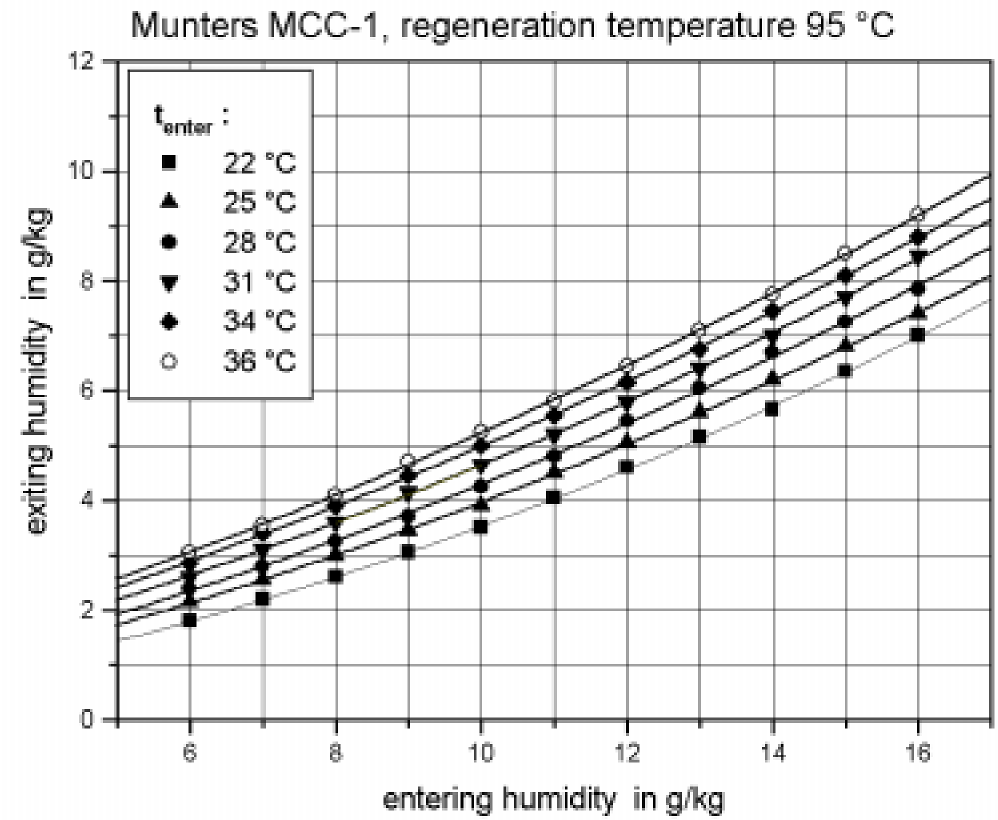

Figure A9 : Humidity ratio leaving the desiccant wheel as a function of the entering humidity ratio and with a regeneration temperature of 95 ^{o}C (manufacturer's data)

17.7 Results

17.7.1 Optimization of the schedules

17.7.1.1 Berlin

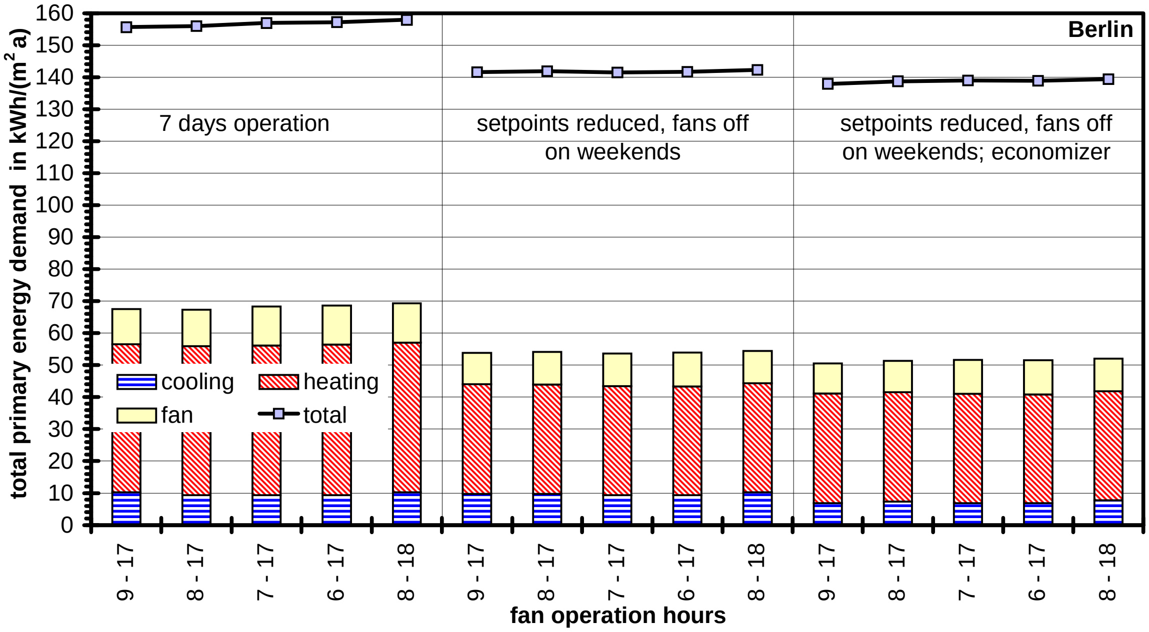

Figure A10 : VAV system : Annual primary energy demand for Berlin in kWh/(m^2 a) versus the operating hours of fans and chillers.

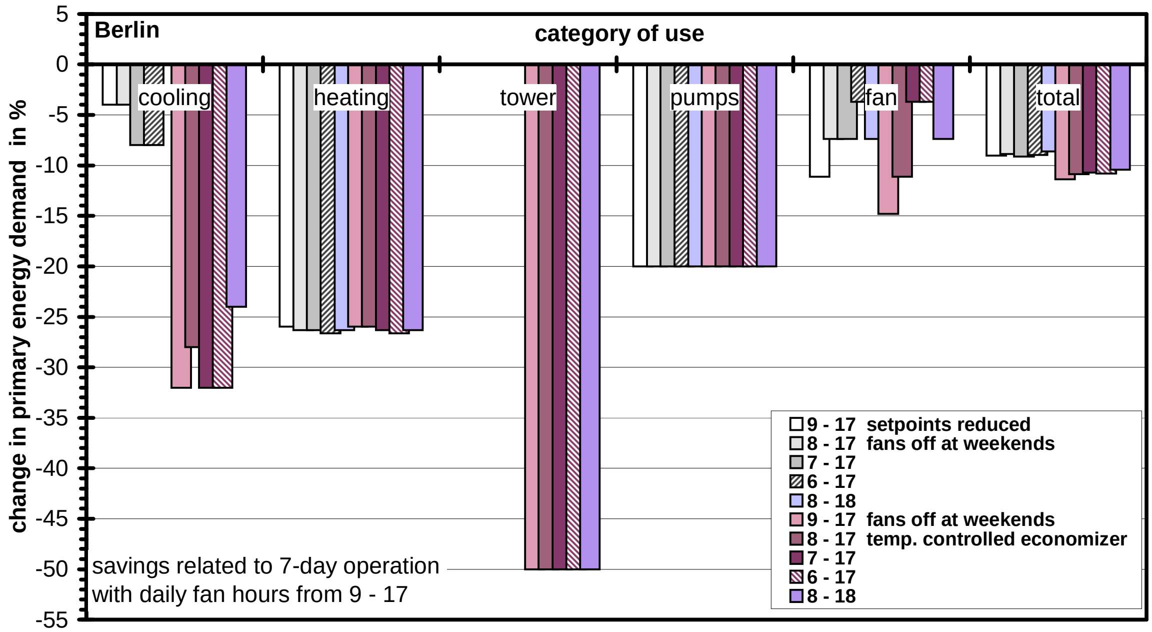

Figure A11 : VAV system : Primary energy savings in % due to different operation strategies and fan running hours for Berlin.

17.7.1.2 Kiel

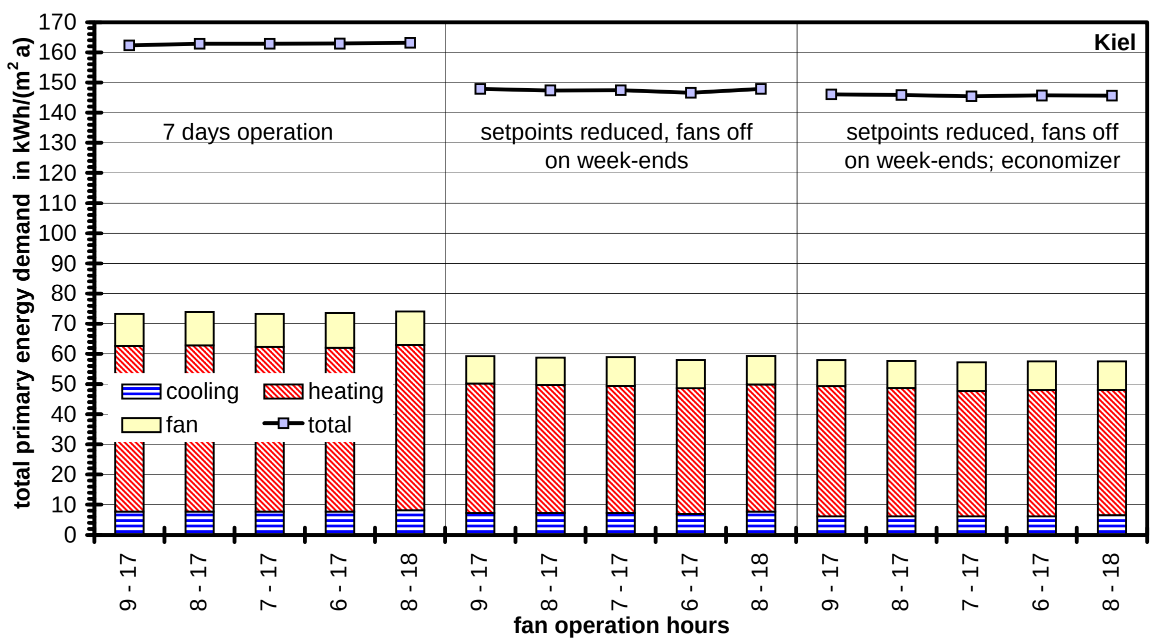

Figure A12 : VAV system : Annual primary energy demand for Kiel in kWh/(m^2 a) versus the operating hours of fans and chillers.

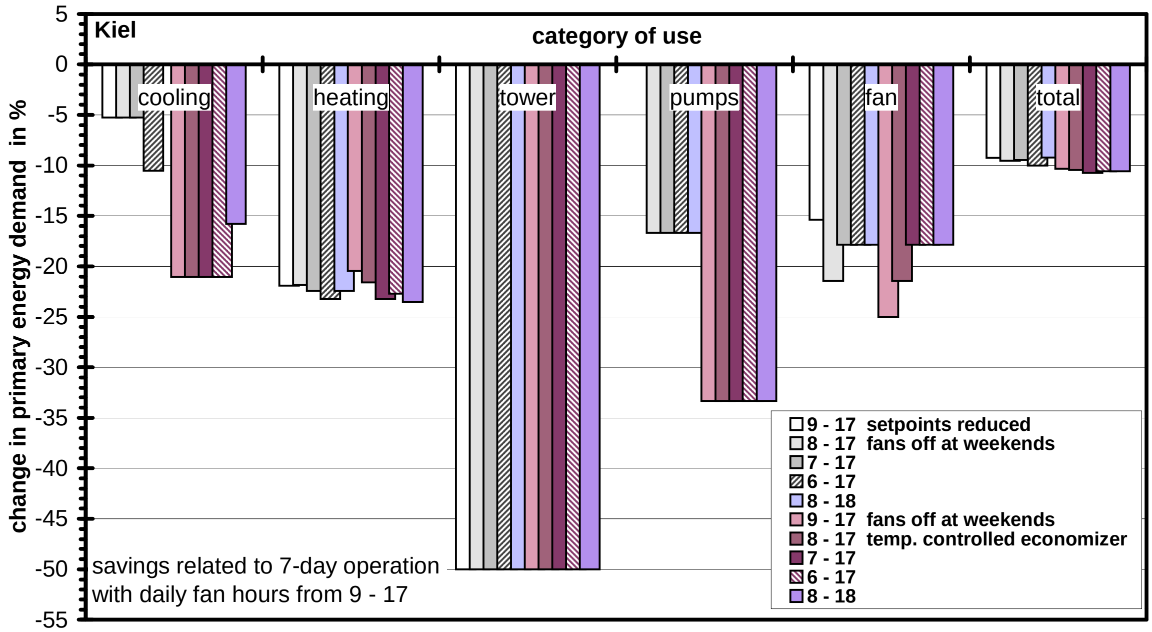

Figure A13 : VAV system : Primary energy savings in % due to different operation strategies and fan running hours for Kiel.

17.7.1.3 Locarno

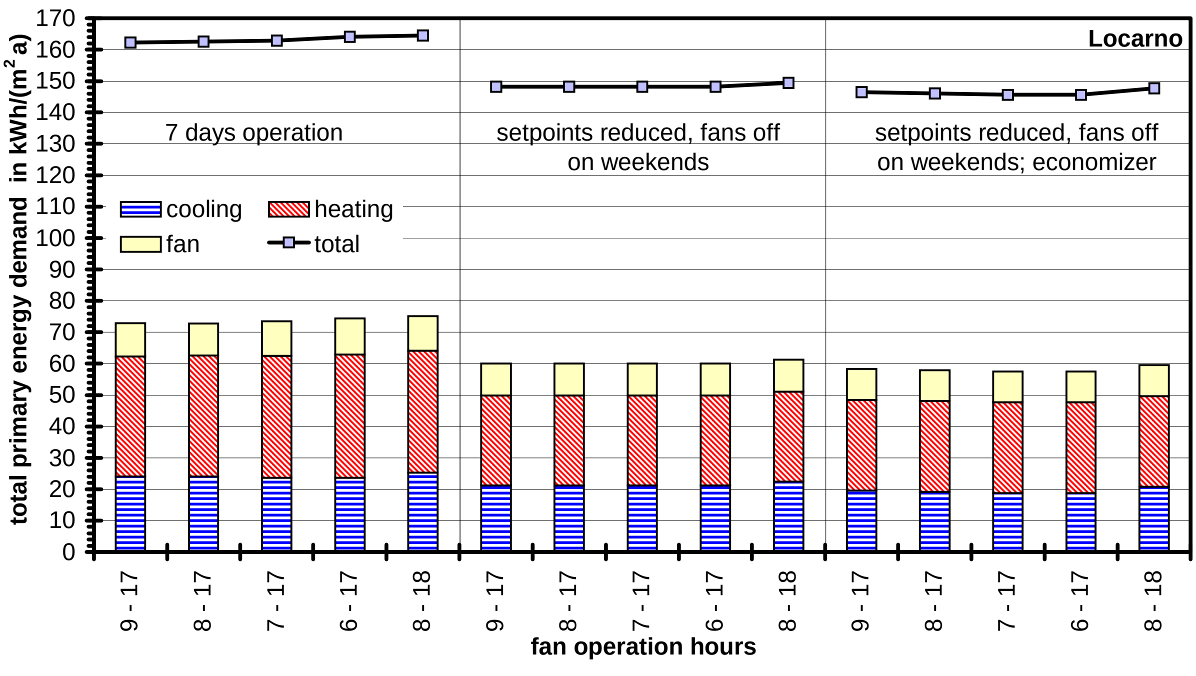

Figure A14 : VAV system : Annual primary energy demand for Locarno in kWh/(m^2 a) versus the operating hours of fans and chillers.

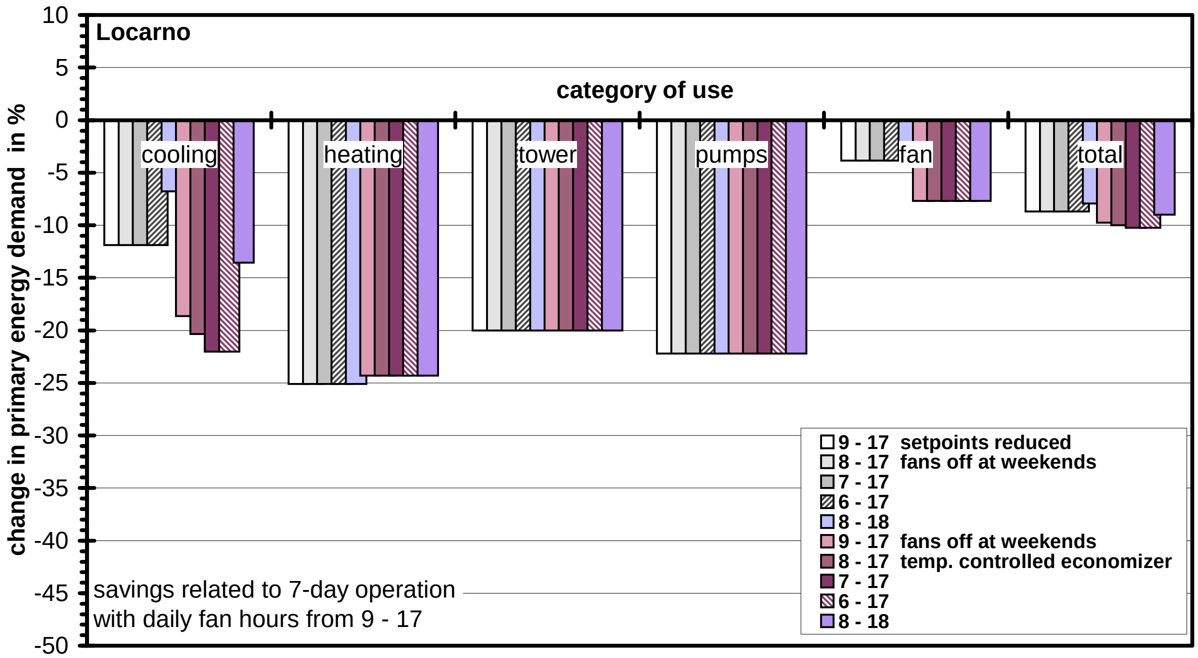

Figure A15 : VAV system : Primary energy savings in % due to different operation strategies and fan running hours for Locarno.

17.7.1.4 San Francisco

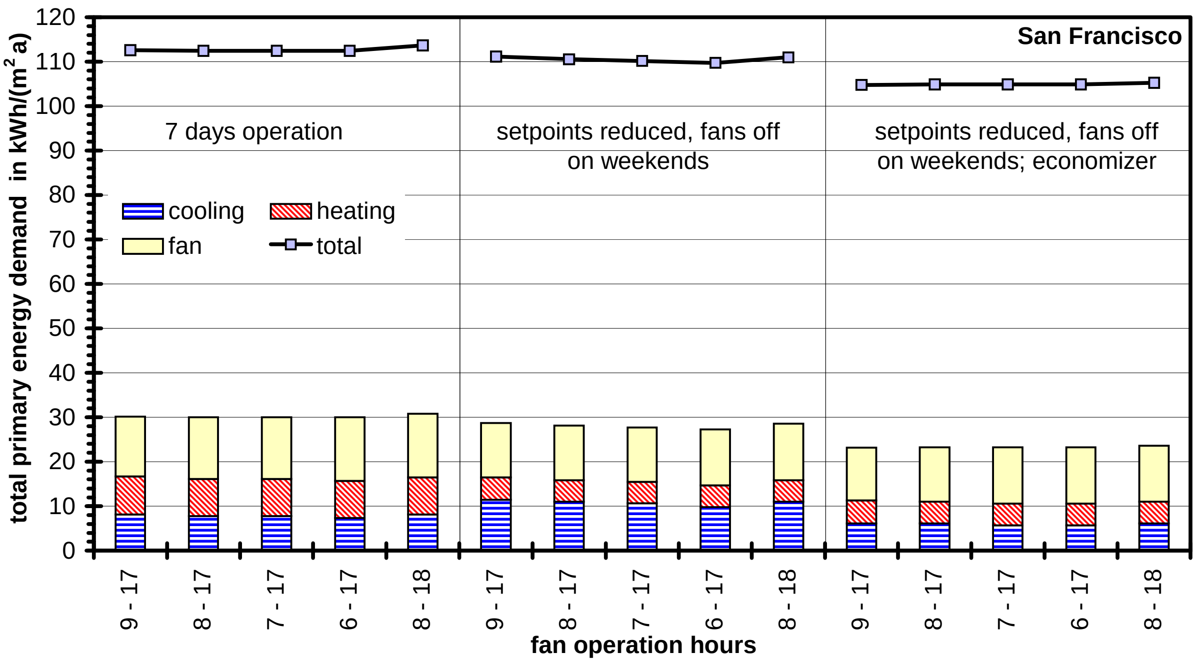

Figure A16 : VAV system : Annual primary energy demand for San Francisco in kWh/(m^2 a) versus the operating hours of fans and chillers.

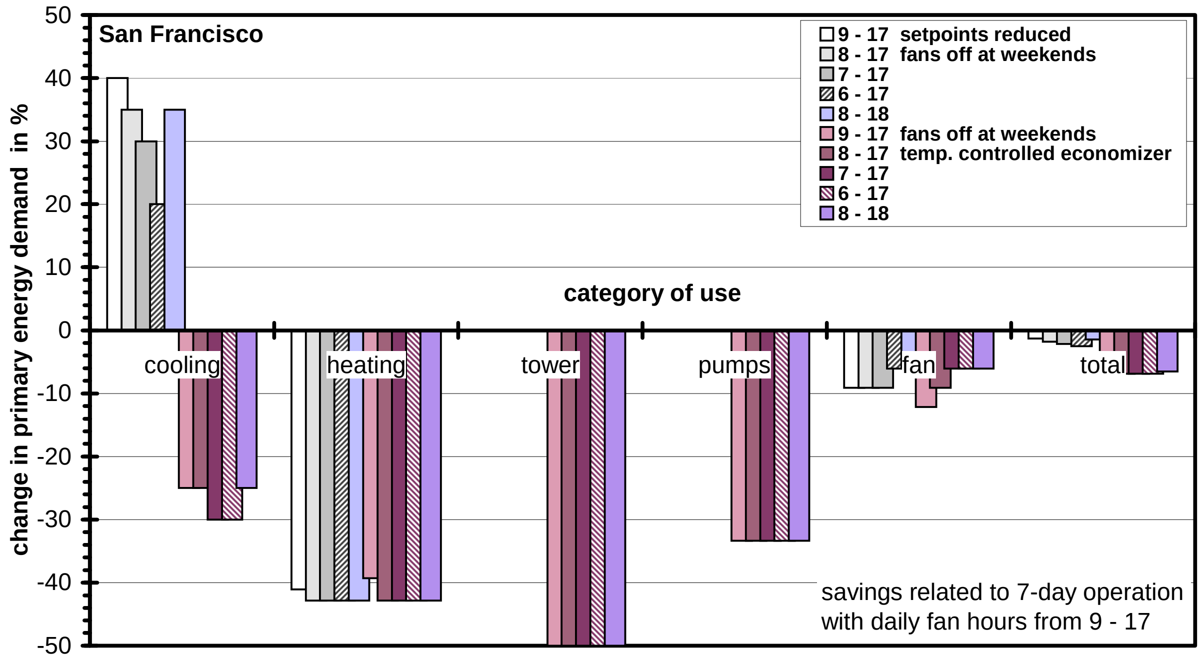

Figure A17 : VAV system : Primary energy savings in % due to different operation strategies and fan running hours for San Francisco.

17.7.2 Natural night ventilation

17.7.2.1 Thermal comfort

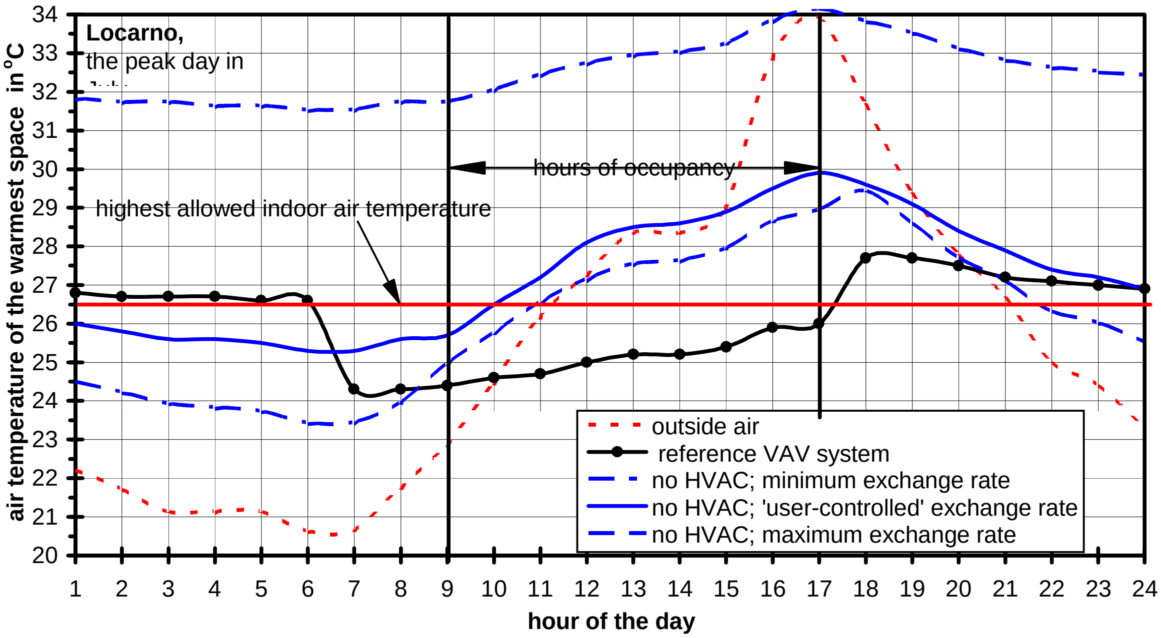

Figure A18 : Night ventilation : Comparison of indoor air temperatures by using natural ventilation with different air exchange rates and the reference VAV system (Locarno).

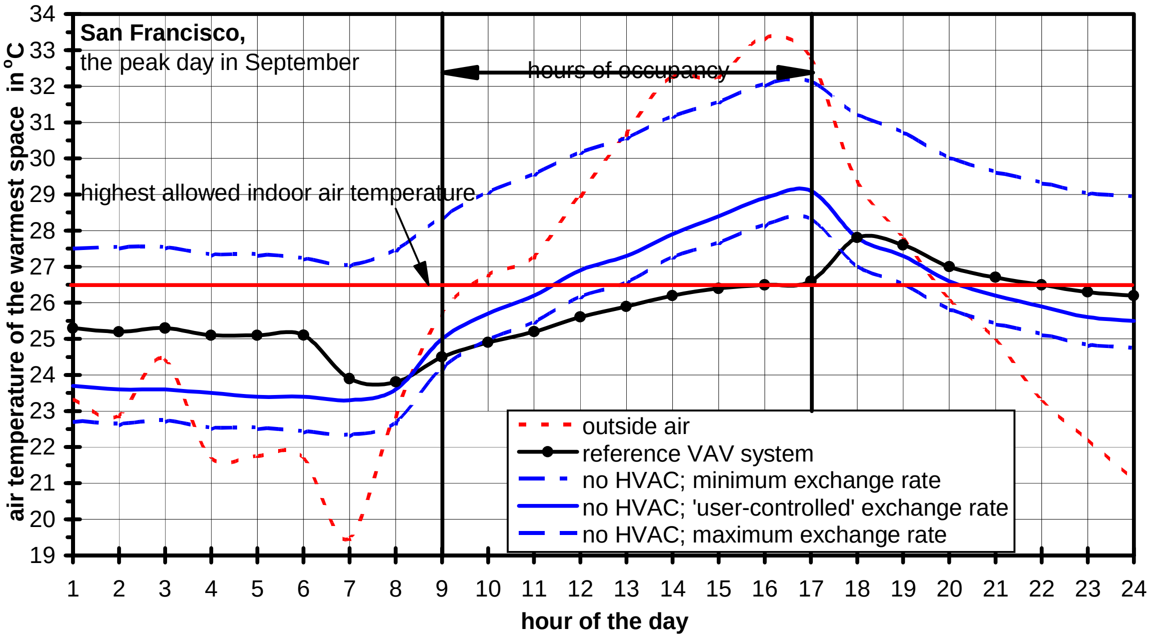

Figure A19 : Night ventilation : Comparison of indoor air temperatures at the peak day by using natural ventilation with different air exchange rates and the reference VAV system (San Francisco).

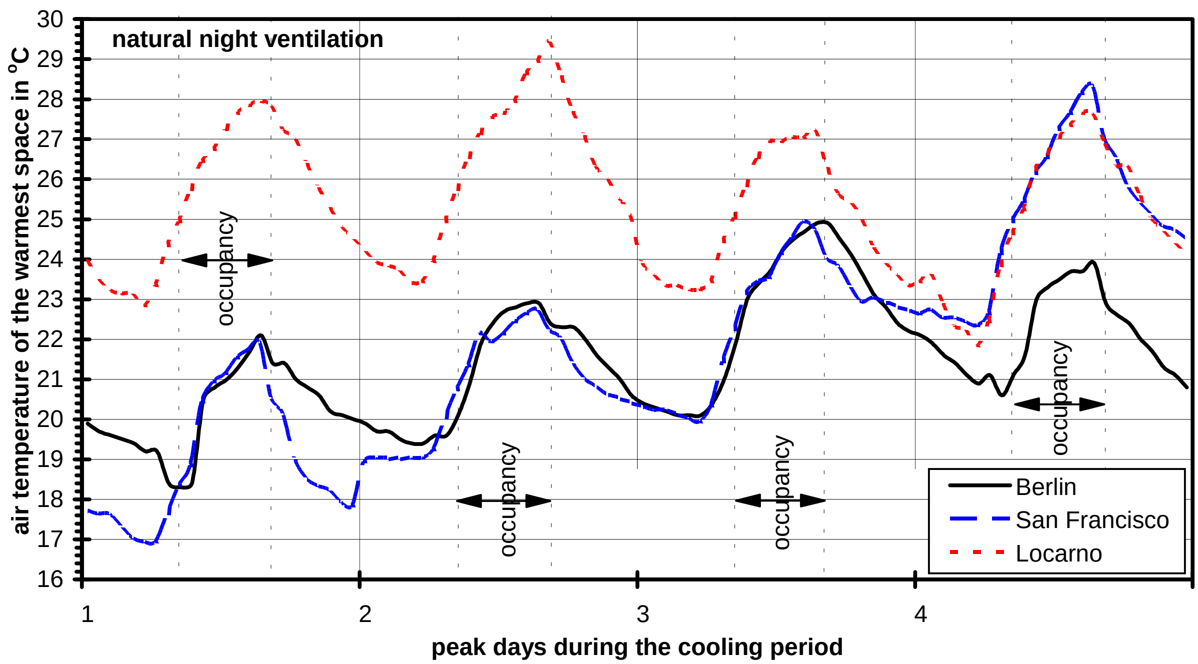

Figure A20 : Night ventilation : Indoor air temperature for Berlin, Locarno and San Francisco at four consecutive days during the cooling peak period when natural night ventilation is used.

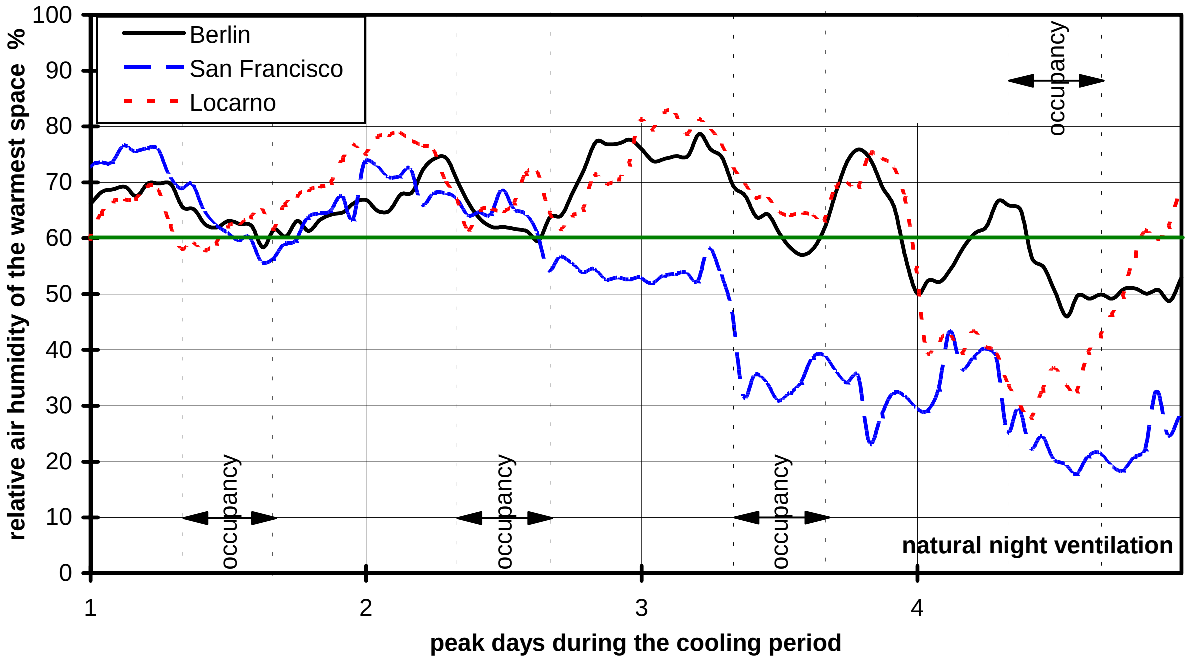

Figure A21 : Night ventilation : Relative indoor air humidity for Berlin, Locarno and San Francisco at four consecutive days during the cooling peak period.

17.7.3 Mechanical night ventilation with compression chiller

17.7.3.1 Locarno

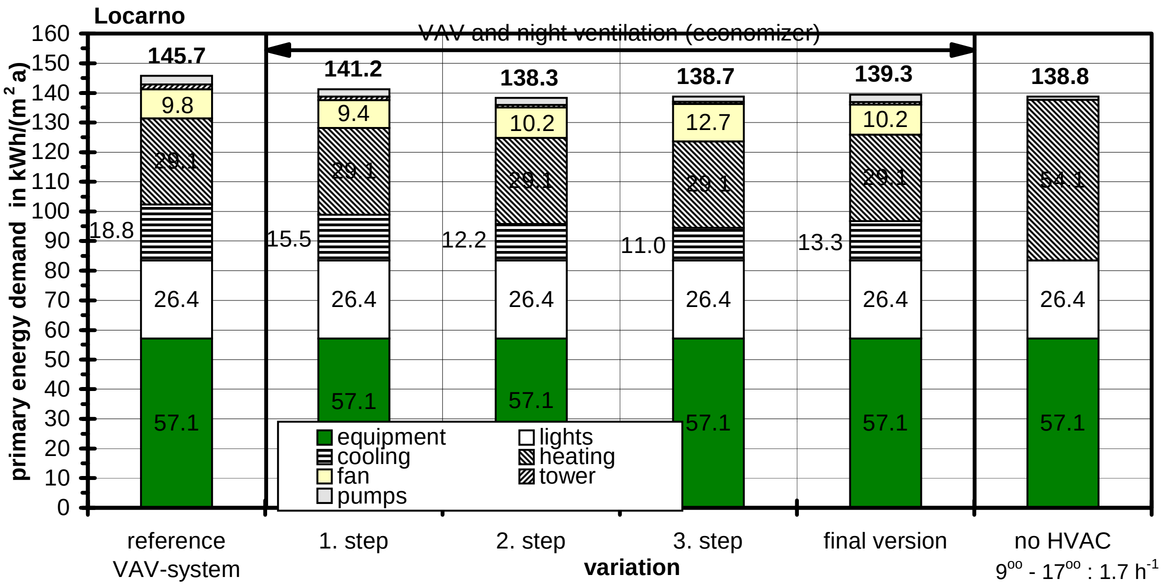

Figure A22 : Night ventilation : Comparison of the primary energy demand of the building located in Locarno using different chillers sizes in addition to mechanical night ventilation.

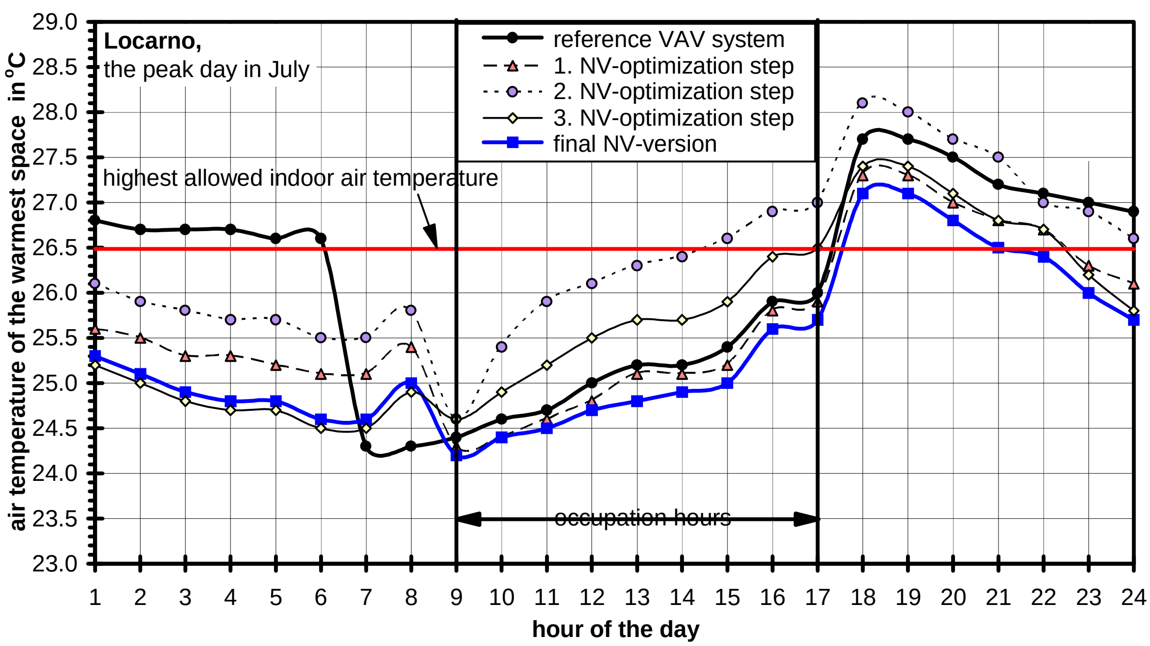

Figure A23 : Night ventilation : Indoor air temperatures for different mechanical night ventilation strategies with chiller in Locarno.

17.7.3.2 San Francisco

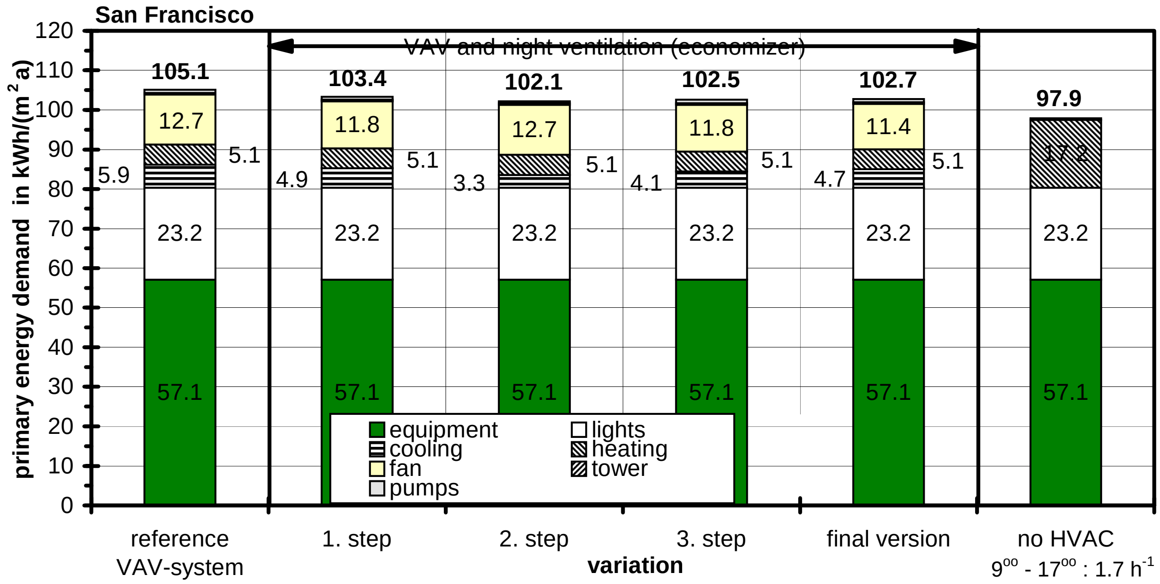

Figure A24 : Night ventilation : Comparison of the primary energy demand of the building located in San Francisco using different chillers sizes in addition to mechanical night ventilation.

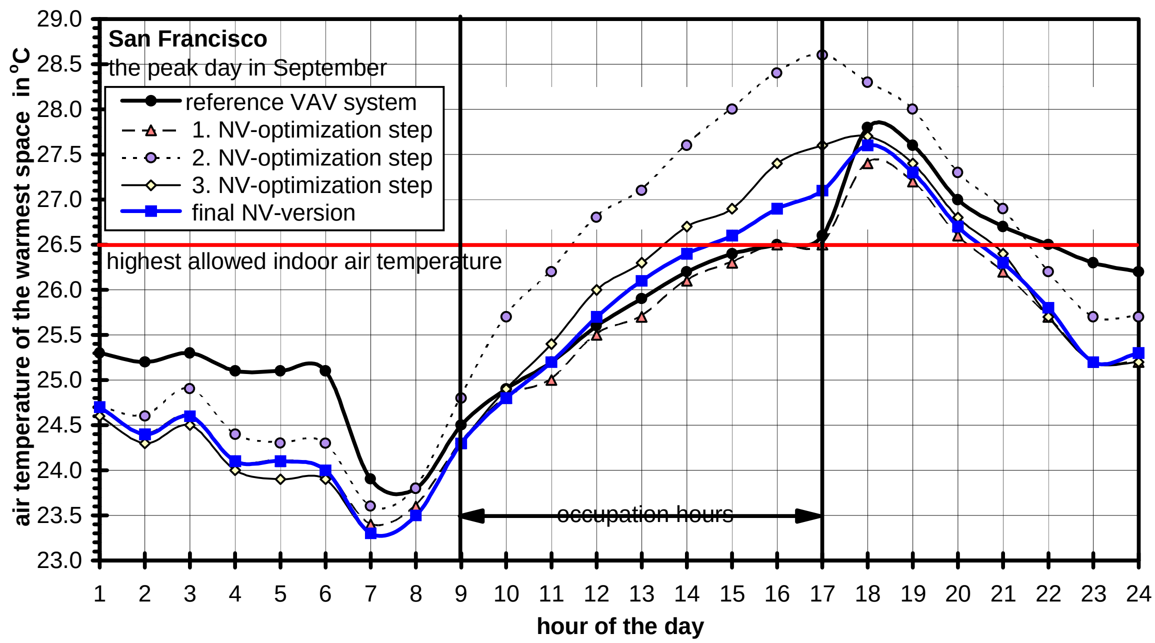

Figure A25 : Night ventilation : Indoor air temperatures for different mechanical night ventilation strategies with compressive cooling in San Francisco.

17.7.3.3 Characteristic data

Table A19 : Night ventilation : Comparison of the characteristic data of the different variations for mechanical night ventilation with compression chiller and the reference VAV system for Berlin, Locarno and San Francisco.

		$\dot{Q}_{chiller}$	$\dot{V}_{fan,max}$	$\dot{V}_{day,max}$	$\dot{V}_{night}$	$Q_{primary}$	
		kW	m^3/h	m^3/h	m^3/h	kWh/m^2	remarks
Berlin							
NV	1. step	20	4000	3900	2000	135.8	still saving potential
	2. step	10	4000	3900	2000	135.0	too warm
	3. step	10	4000	3900	4800	138.3	too much energy
	final version	**10**	**4800**	**3900**	**2900**	**134.6**	
VAV system		27	4800	4700	-	138.7	
Locarno							
NV	1. step	30	6900	6500	3500	141.2	still saving potential
	2. step	15	6900	6800	3500	138.3	too warm
	3. step	15	7700	7400	6400	138.7	too humid
	final version	**20**	**7700**	**7400**	**4800**	**139.5**	
VAV system		39	7700	7400	-	145.6	
San Francisco							
NV	1. step	15	4900	4800	2400	103.2	still saving potential
	2. step	7.5	4900	4900	2400	102.0	too warm
	3. step	10	5900	5500	2900	102.4	still too warm
	final version	**12**	**5900**	**5400**	**2300**	**102.4**	
VAV system		16	5900	5700	-	104.9	

$\dot{V}$: maximum fan capacity, the pressure drop value of the duct system set in DOE-2E refers to this airflow.

17.7.4 Mechanical night ventilation without additional cooling

17.7.4.1 Locarno

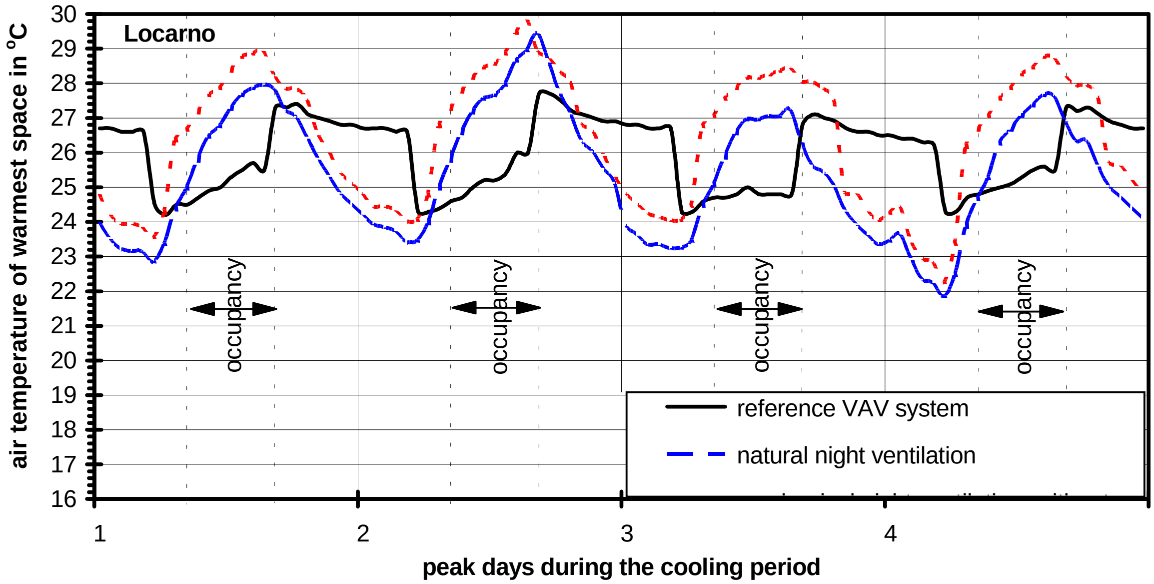

Figure A26 : Night ventilation : Indoor air temperatures during a cooling peak period for night ventilation strategies without chiller compared with those with the reference VAV system (Locarno).

17.7.4.2 San Francisco

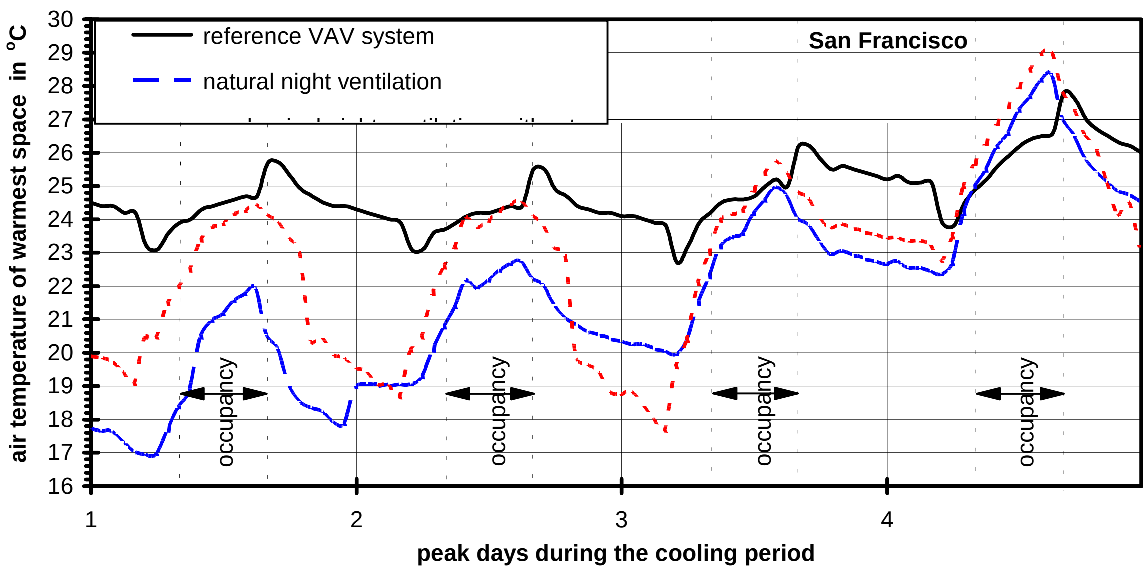

Figure A27 : Night ventilation : Indoor air temperatures during a cooling peak period for night ventilation strategies without chiller compared with those with the reference VAV system (San Francisco).

17.7.4.3 Characteristic data

Table A20 : Night ventilation : Characteristic data of the variations with mechanical night ventilation without additional cooling for Berlin, Locarno and San Francisco.

		$\dot{Q}_{chiller}$	$\dot{V}_{day,max}$	$\dot{V}_{night}$	$Q_{primary}$
		kW	m^3/h	m^3/h	kWh/m^2
Berlin	**NV without chiller**	-	2500	8400	139.5
	VAV system	27	4700	-	138.7
Locarno	**NV without chiller**	-	2500	15300	145.2
	VAV system	39	7400	-	145.6
San Francisco	**NV without chiller**	-	2500	10300	106.9
	VAV system	16	5400	-	104.9

Table A21 : Annual energy costs and primary energy consumption of the HVAC system for different cooling strategies and climates. The reference VAV system with a compression chiller (COP = 3.0) represents 100 % for both categories.

cooling strategy		TRY 3 3)	Locarno	Red Bluff	San Francisco
reference VAV system	$Q_{a,Pr}$	100	100	100	100
	$P_{a,Pr}$	100	100	100	100
VAV system (COP = 4.0)	$Q_{a,Pr}$	97	92	87	94
	$P_{a,Pr}$	96	92	86	93
radiant cooling system	$Q_{a,Pr}$	96	90	70	87
	$P_{a,Pr}$	95	89	67	85
natural night ventilation	$Q_{a,Pr}$	127	--1)	--2)	72
	$P_{a,Pr}$	104	--1)	--2)	47
mech. night ventilation	$Q_{a,Pr}$	100	--1)	--2)	107
	$P_{a,Pr}$	101	--1)	--2)	108
night and compr. cooling	$Q_{a,Pr}$	92	90	90	90
	$P_{a,Pr}$	91	89	89	89
evaporative cooling	$Q_{a,Pr}$	--1 + 2)	--1 + 2)	75	77
	$P_{a,Pr}$	--1 + 2)	--1 + 2)	73	74
evap. and compr. cooling	$Q_{a,Pr}$	96	89	50	--2)
	$P_{a,Pr}$	95	88	46	--2)
desiccant cooling (DEC)	$Q_{a,Pr}$	97	92	--2)	--2)
	$P_{a,Pr}$	97	83	--2)	--2)
DEC with district heat 6)	$Q_{a,Pr}$	92	75	--2)	--2)
	$P_{a,Pr}$	98	82	--2)	--2)
DEC with waste heat 4)	$Q_{a,Pr}$	89	68	--2)	--2)
	$P_{a,Pr}$	92	68	--2)	--2)
single-stage abs. cooling	$Q_{a,Pr}$	111	123	131	121
	$P_{a,Pr}$	102	103	98	104
dual-effect abs. cooling	$Q_{a,Pr}$	102	104	103	106
	$P_{a,Pr}$	97	92	82	95
abs.cooling 5)+ district heat 6)	$Q_{a,Pr}$	98	93	85	98
	$P_{a,Pr}$	102	101	115	114
abs.cooling 5)+ waste heat 4)	$Q_{a,Pr}$	91	77	61	85
	$P_{a,Pr}$	89	74	57	83

1) This variation can not provide thermal comfort completely.
2) This variation has not been investigated.
3) These numbers can also be used as estimate for Kiel (German TRY 1).
4) Waste heat is assumed to be for free and no primary energy consumption is involved.
5) Single-stage absorption chiller.
6) 0.38 kWh primary energy is consumed to provide 1 kWh district heat.